AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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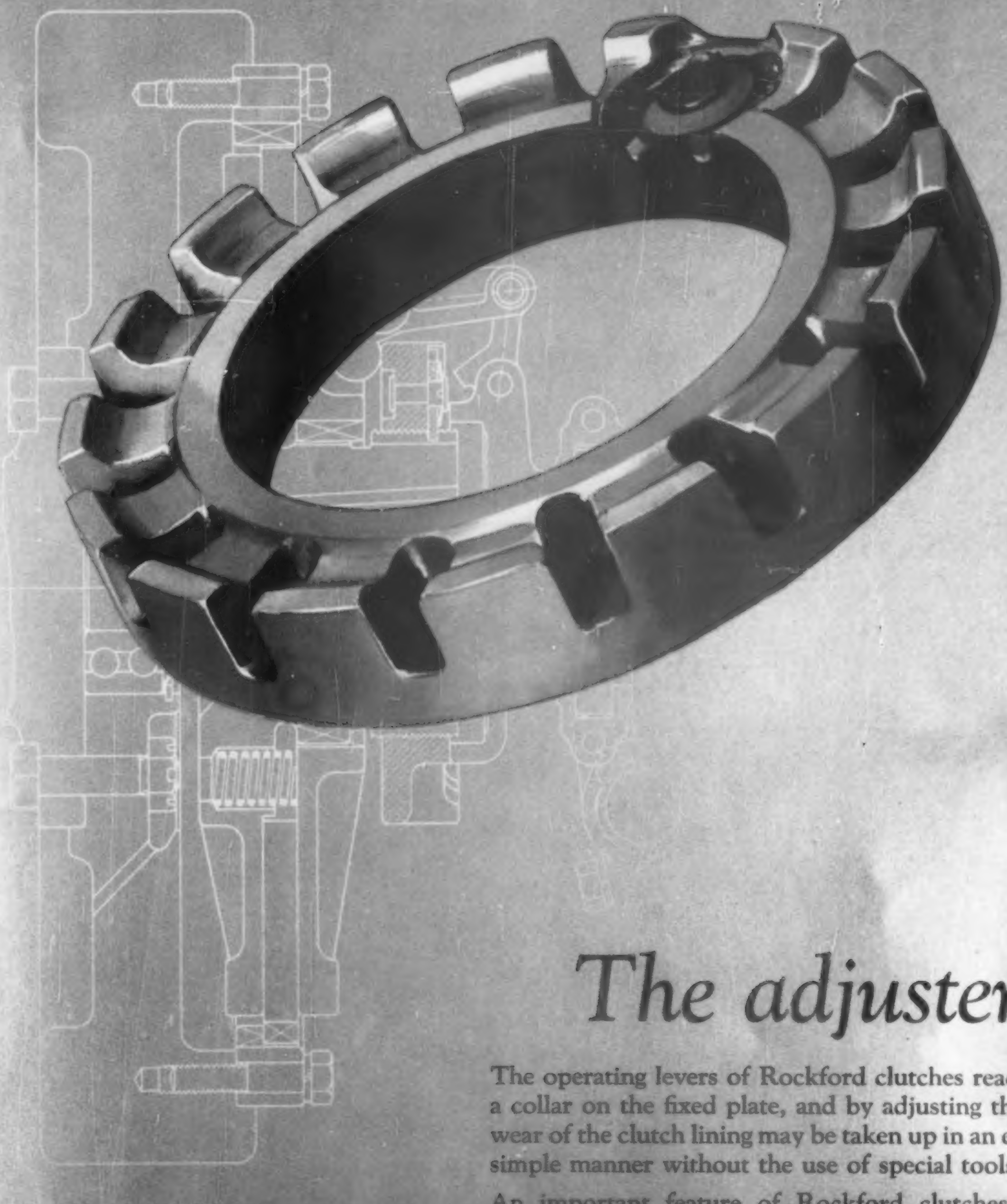


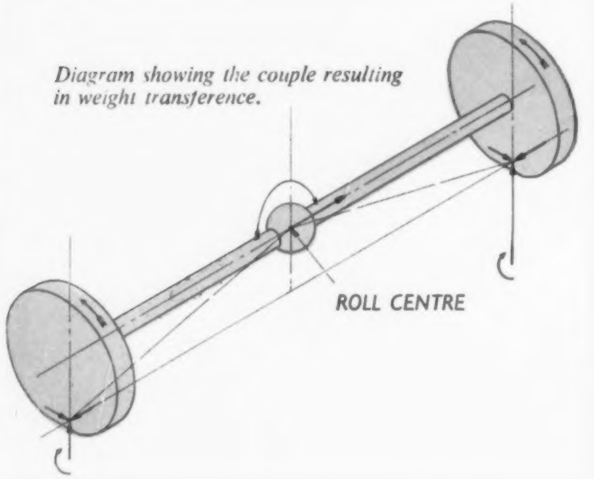

PATENTS PENDING
AUTOMOTIVE PRODUCTS COMPANY LIMITED

MAKING STEERING LIGHTER

Weight transference in cornering

We have seen that as far as parking torque is concerned, i.e. heaviness of steering in low-speed manoeuvring, we can improve it mechanically by providing minimum friction in the steering joints and pivots and maximum efficiency of the steering box, and as far as the tyre is concerned—by increasing the pressure. When the car is on the move, the tyres produce self-aligning torque

Diagram showing the couple resulting in weight transference.



and drag-difference torque as a result of the drift angles induced by cornering, and we are now concerned with the effect on these torques of the way in which the weight transference from inner to outer wheels on a corner is shared between front and rear.

Perhaps the best way we can study this for our present purpose is to take an assumed pair of wheels developing a given sideways force and increase the weight transference by stages and see what effect this increase has on the total self-aligning torque from all sources. For this purpose we take again the tyre properties given in the I.Mech.E., A.D., paper "Tyre Characteristics as Applicable to Vehicle Stability Problems", by Hartley & Joy. For an initial load per wheel of 950 lbs. we assume a total cornering force per pair of wheels of 800 lbs. The drift angle of the pair of wheels rises from 4·4° at zero weight transference to 5·4° at 500 lbs. weight transference. The self-aligning torque of the tyres rises from 77 lbs./ft. at zero weight transference to 98 lbs./ft. at 500 lbs. weight transference. With the assumed offset of the wheel plane at the rotational axis of 3" from the king pin, the drag difference effect rises from zero at zero weight transference to 11 lbs./ft. at 500 lbs. weight transference. We have therefore added just over 40% to the total tyre self-aligning torque by our weight transfer of 500 lbs.

Increase of weight transfer at the front is a means of increasing understeer tendency, as the increase in drift angle noted above will imply; we cannot therefore even consider the increase of tyre pressure as a cure of heaviness resulting from increased weight transfer, as the increased tyre pressure would reduce the drift angle which we have been at pains to put up by increasing the weight transference. We can help ourselves a little by various means, but they can only make small differences. We can, for instance, eliminate the Ackermann effect from the steering linkage. This implies that the outside wheel on a corner turns through a greater angle and as the weight is transferred to this wheel on a corner, effectively some extra steering angle is obtained; this however also has an oversteering tendency and may be impossible or at least undesirable because of that.

We can also get a little of our understeer by roll steer effect. It is unwise to overdo this, however, as otherwise the handling behaviour on a straight road with the car unrolled will be considerably different from the cornering handling, and one of the reasons for liking understeer is that it makes the car so much easier to control on a straight road.

Basically, heavier dynamic steering is one of the penalties of understeer, and if we like understeer we must put up with some degree of heaviness as a result.

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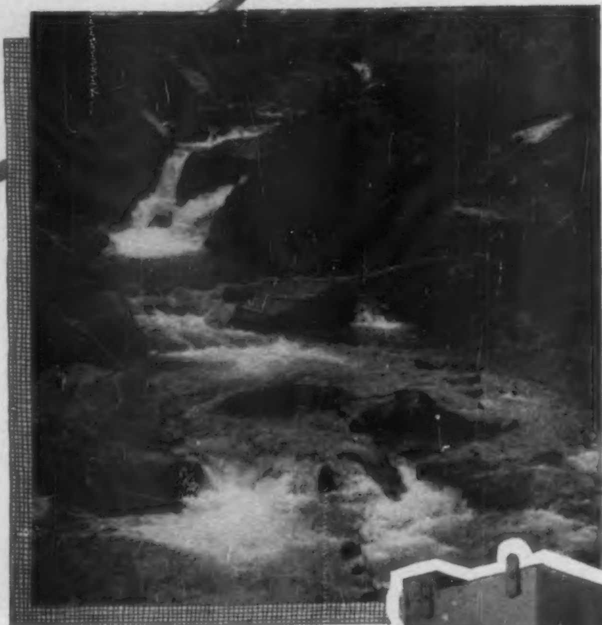

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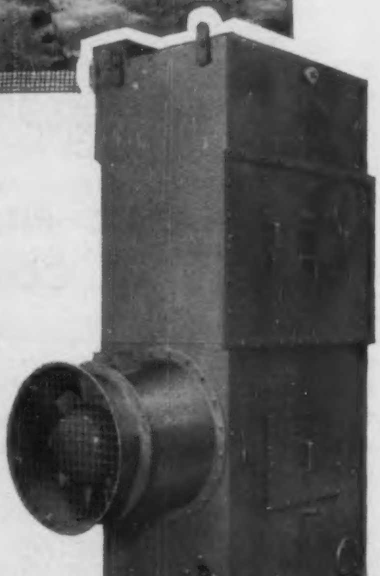

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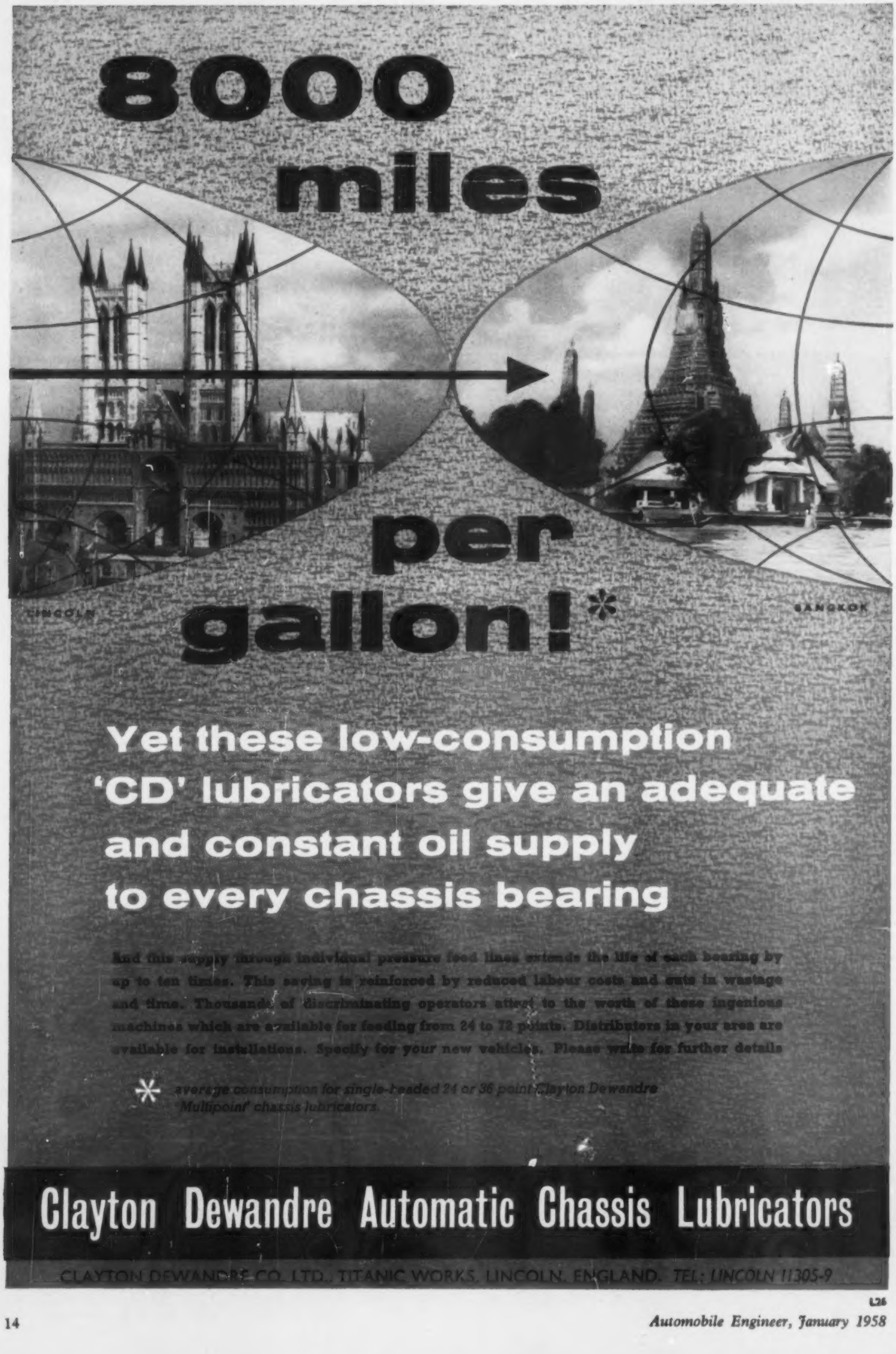

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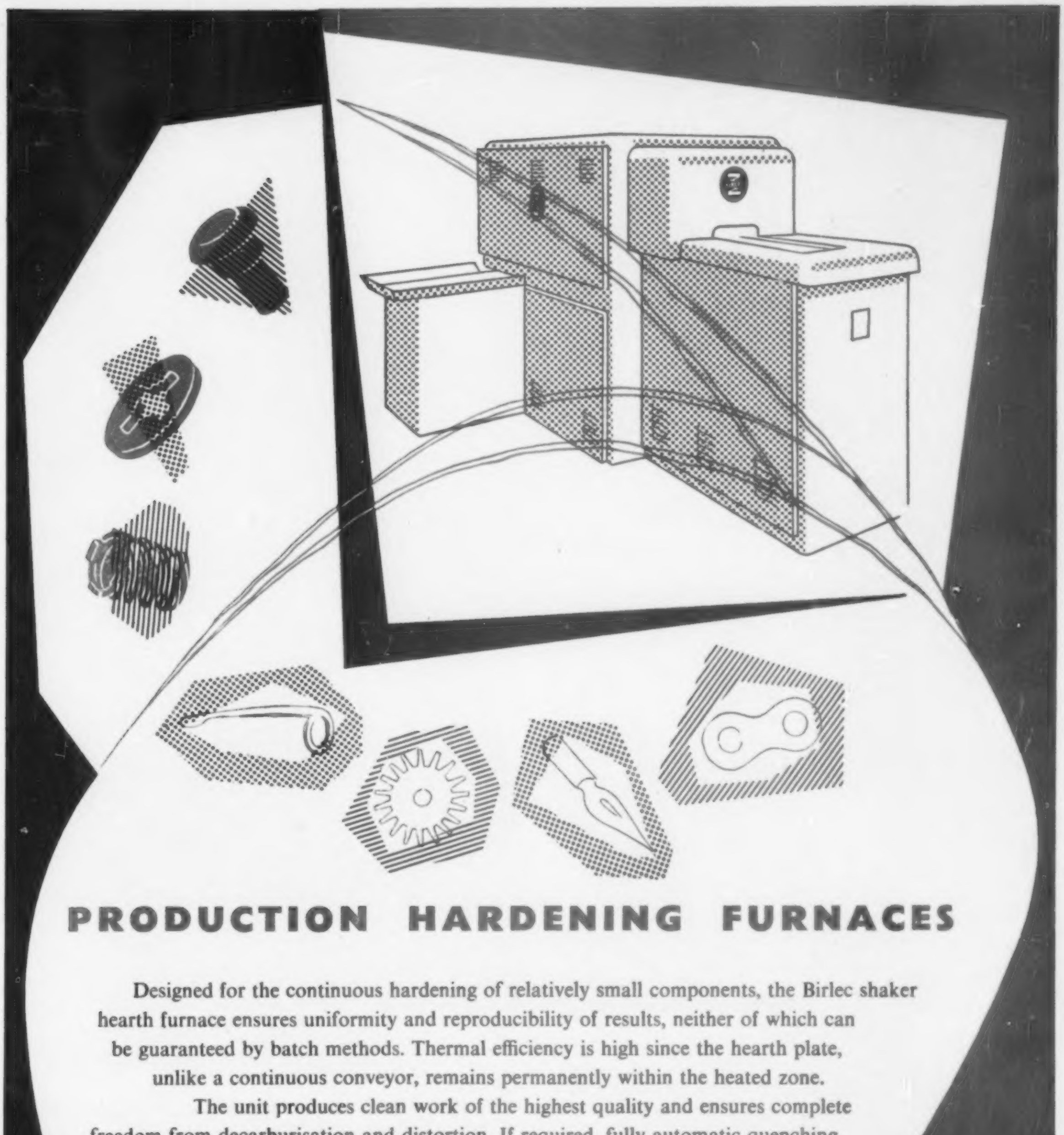

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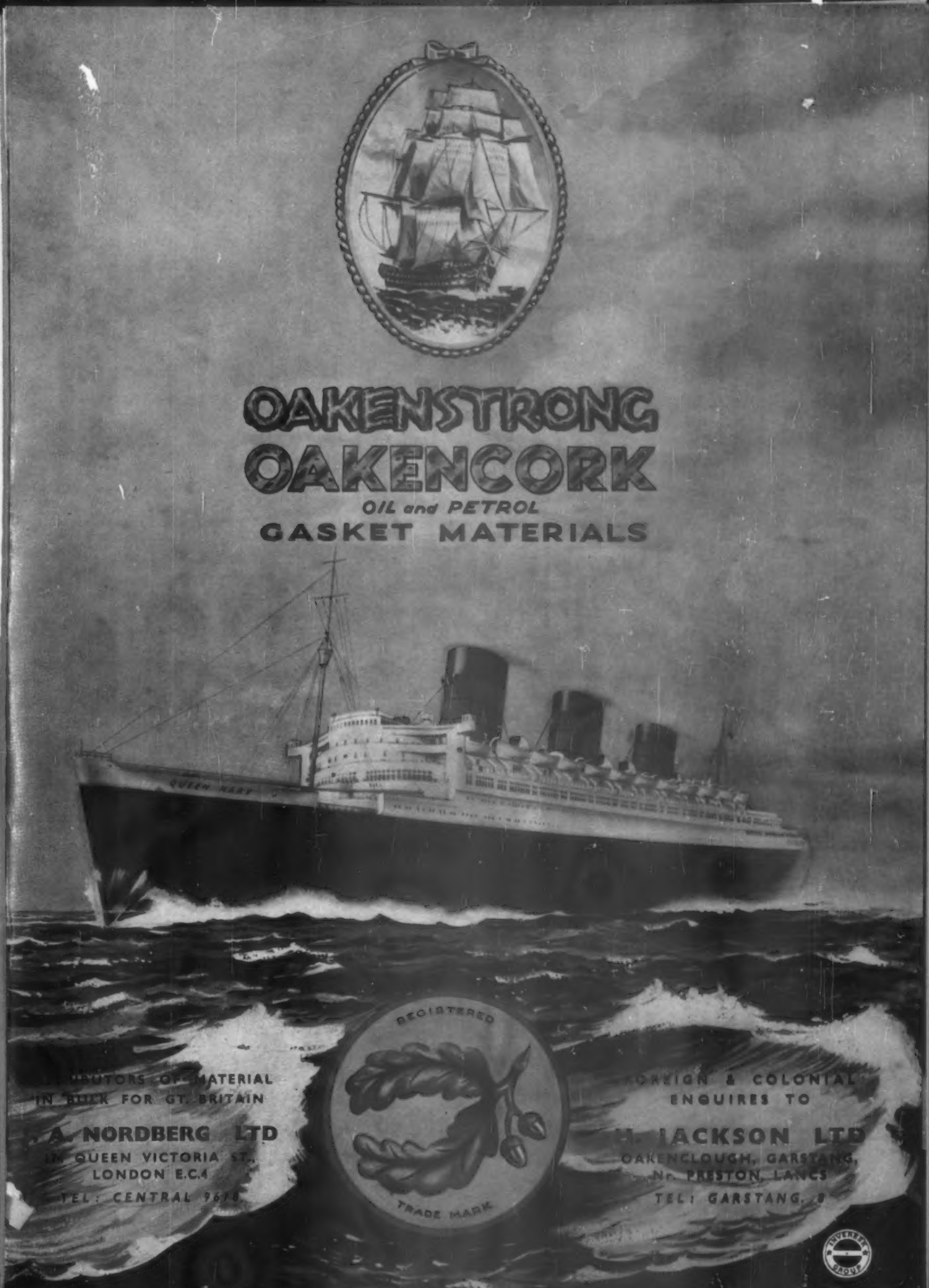
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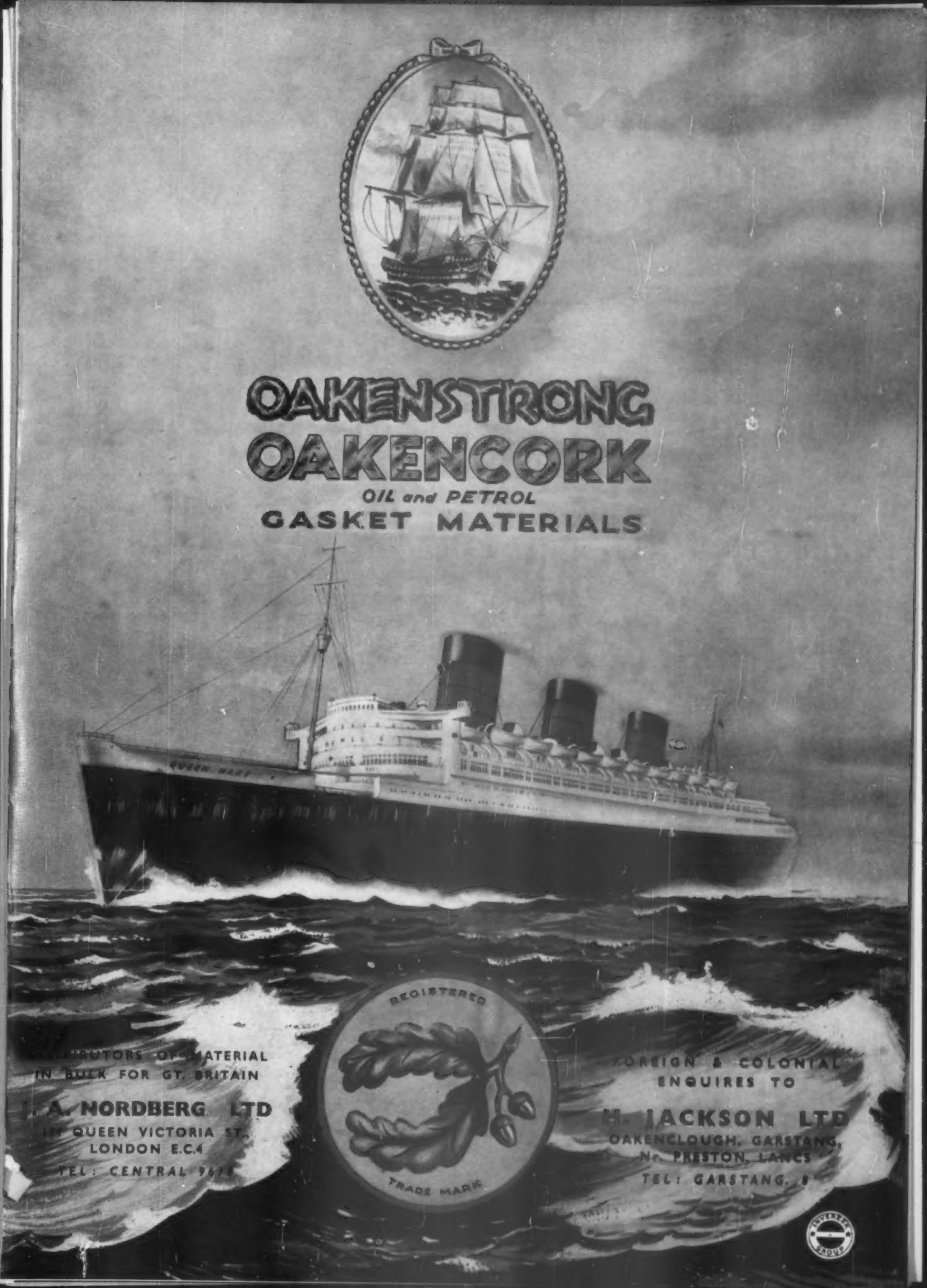
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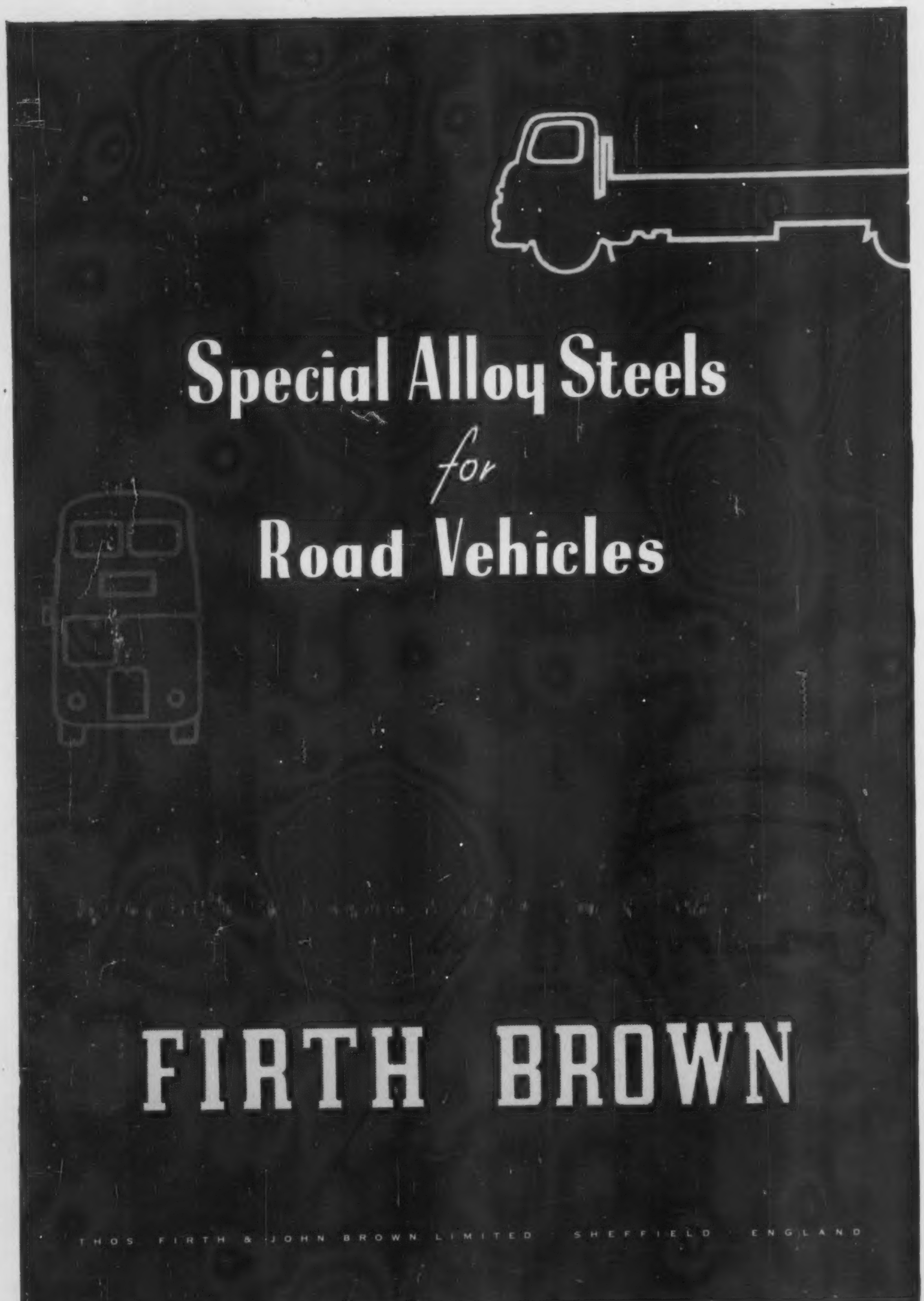
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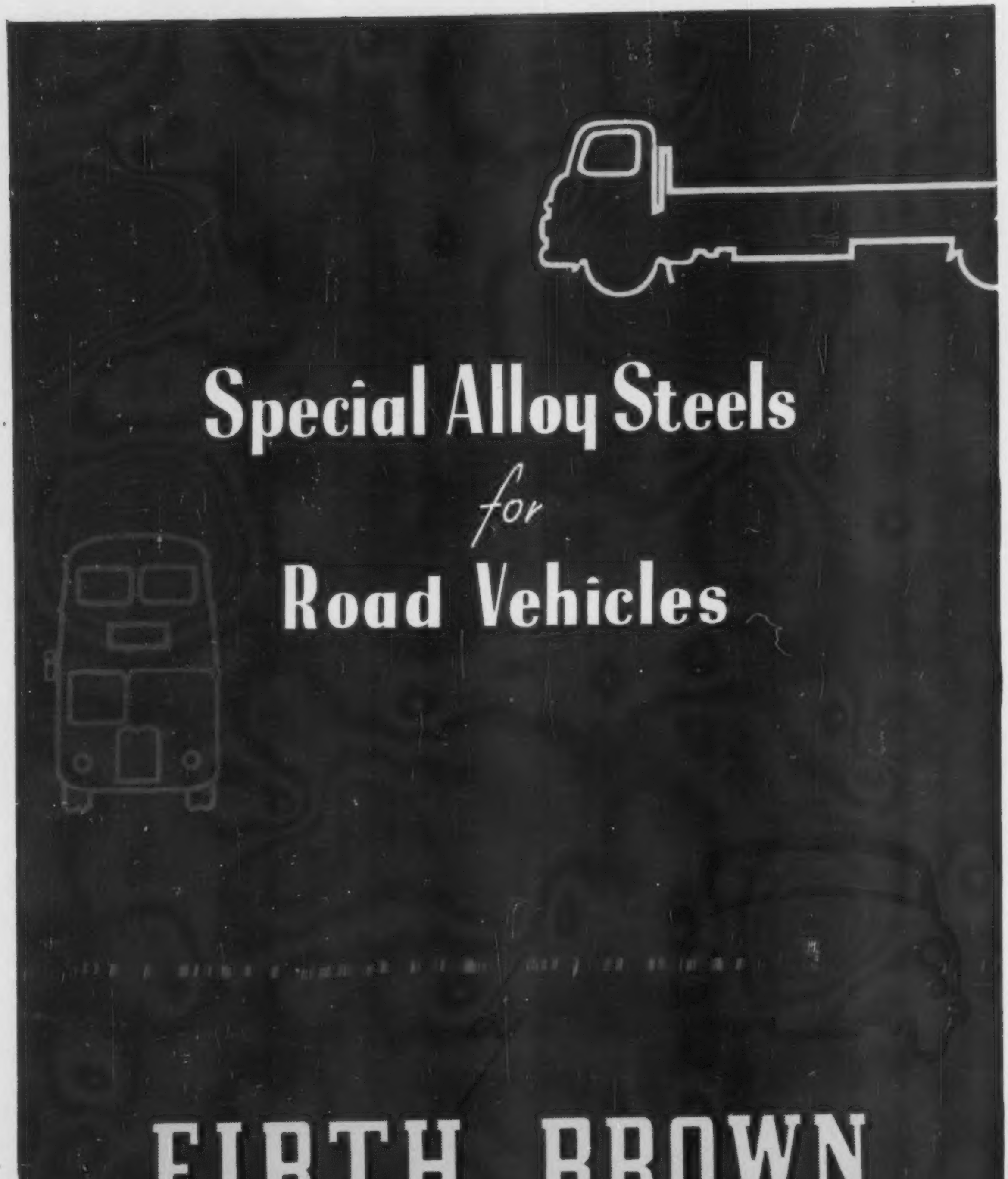
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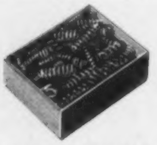

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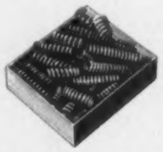

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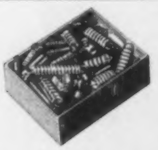

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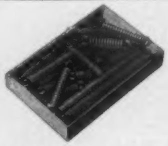

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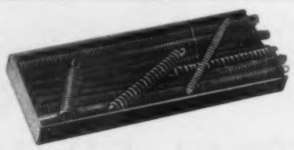

No. 1024. 20 Compression Springs 12″ long. [illegible] to [illegible] diam., 24G to 18G, suitable for cutting into shorter lengths; and 30 Expansions 1[illegible] to 12″ long, 5/32″ to [illegible] diam., 22G to 16G. 24/- each.

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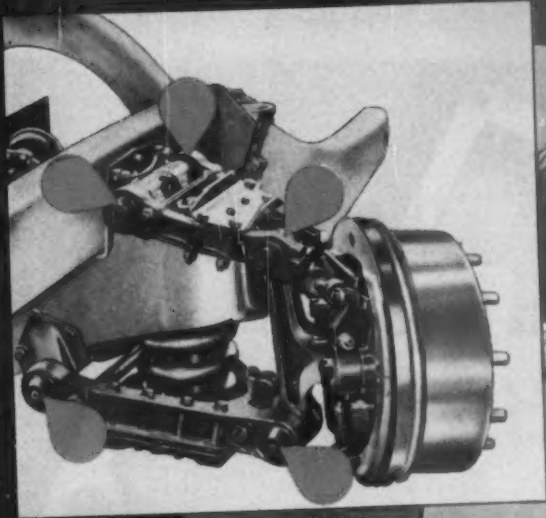
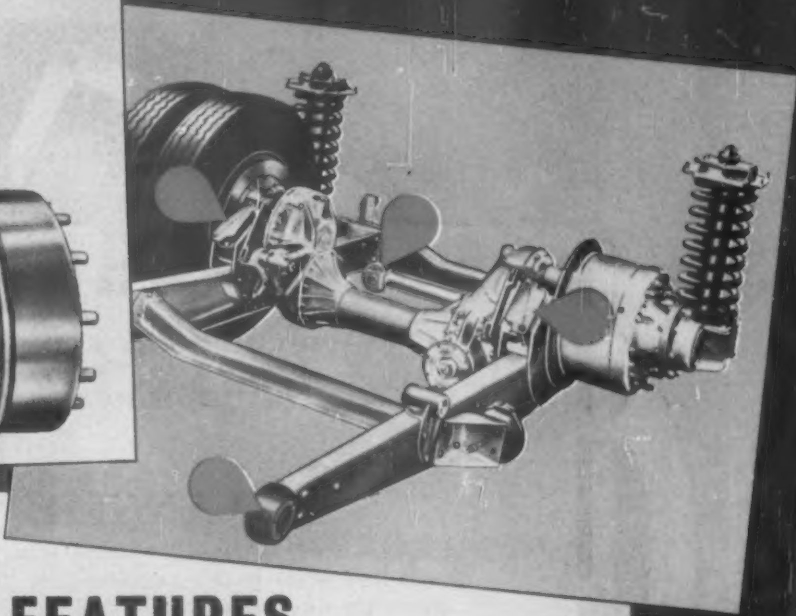
HT21





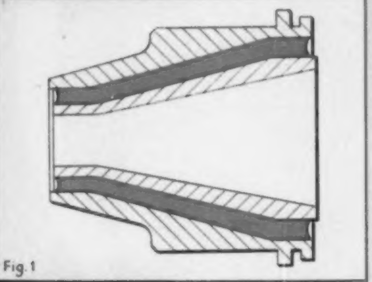


Fig. 1

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The wishbone pivots have Metalastik heavy-duty conical bushes, as shown in the sectional drawing (Fig. 1). These are an important factor in making this type of suspension practicable on a heavy vehicle.

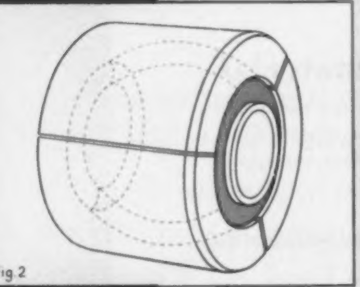

Fig. 2

On both vehicles the rear suspension has two substantial trailing arms pivoted to the main suspension members on Spherilastik self-aligning bearings (Fig. 2). The axle assembly is mounted upon the trailing arms by Metalastik sandwich mountings (Fig. 3) at each end. The mountings are arranged in an inverted V formation and permit relative movement of the wheels while increasing roll stiffness and resisting axle wind-up. The transmission of vibration and noise to the structure is reduced by the mountings.

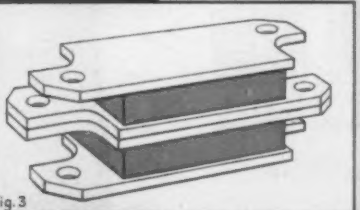

Fig. 3

A Panhard rod with a Spherilastik bearing at one end and an Ultra-Duty bush at the other is fitted.

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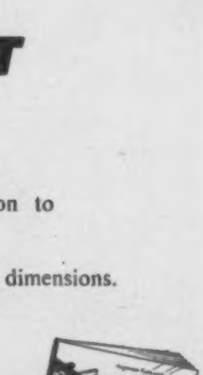

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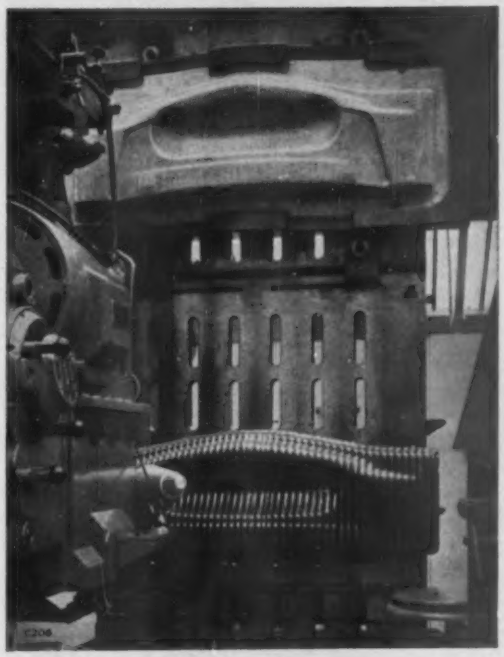

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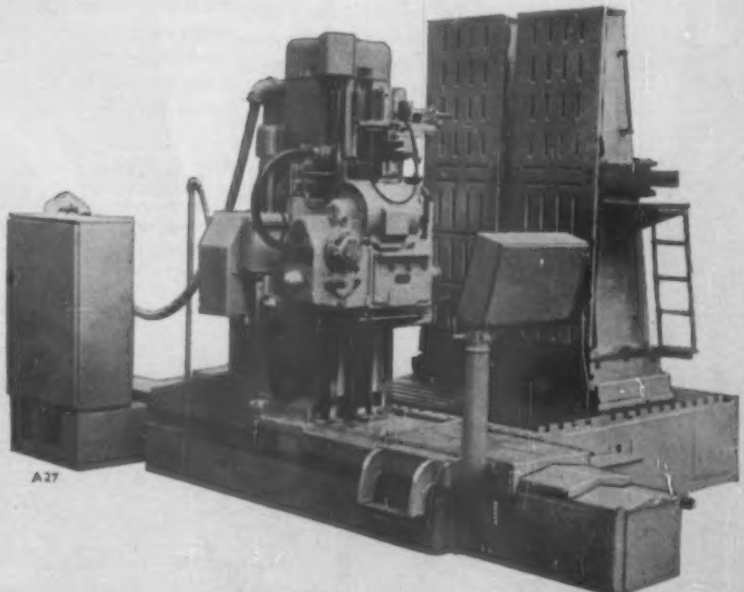

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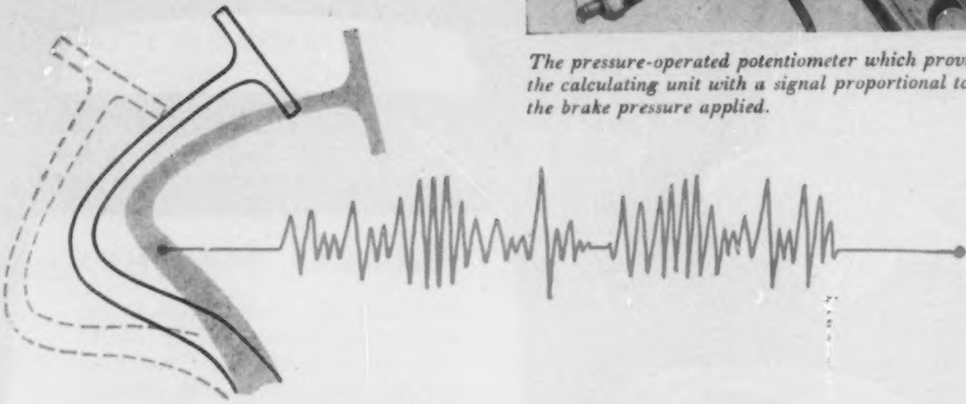
BACKGROUND TO BRAKING No. 1

The pressure-operated potentiometer which provides the calculating unit with a signal proportional to the brake pressure applied.



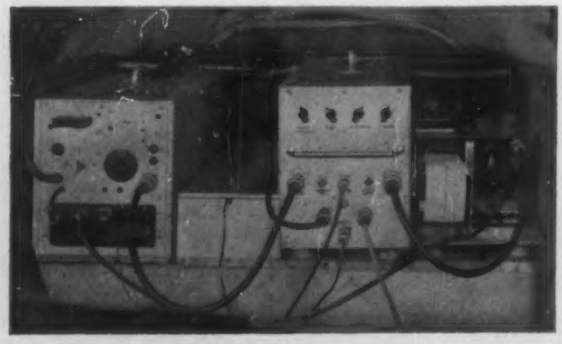
Electronics aid the search for Safety

In devising schedules for tests, Ferodo research workers must know what is expected of a brake lining under practical conditions. Without this information it is possible to overrate the lining—make it break down in a manner which would not occur in service.

The performance of a brake lining depends mainly upon the rate at which work is done at the brake and upon the temperature of the drum surface. To record this information Ferodo technicians have installed ingenious apparatus on a vehicle of the Ferodo Test Fleet.

A tachometer generator, driven by the vehicle's propellor shaft provides a signal which is used to record the speed and the deceleration of the vehicle on a high-speed multi-pen recorder. A voltage proportional to the product of the hydraulic pressure in the brake system and the speed signal is applied to a third pen. Subject to certain necessary precautions, in particular that the brake factor should be frequently checked experimentally, this product is a measurement of the rate of working at the brakes. A thermister is soldered to the surface of the drum to record temperature on a fourth pen.

The results enable Ferodo to devise testing schedules that are accurate and reliable and so to produce brake linings with a high resistance to fade and wear.



FERODO

ANTI-FADE Brake Linings

Some of the apparatus in the back of a test car. The power pack is on the left and the chart can be seen emerging from the four pen recorder.

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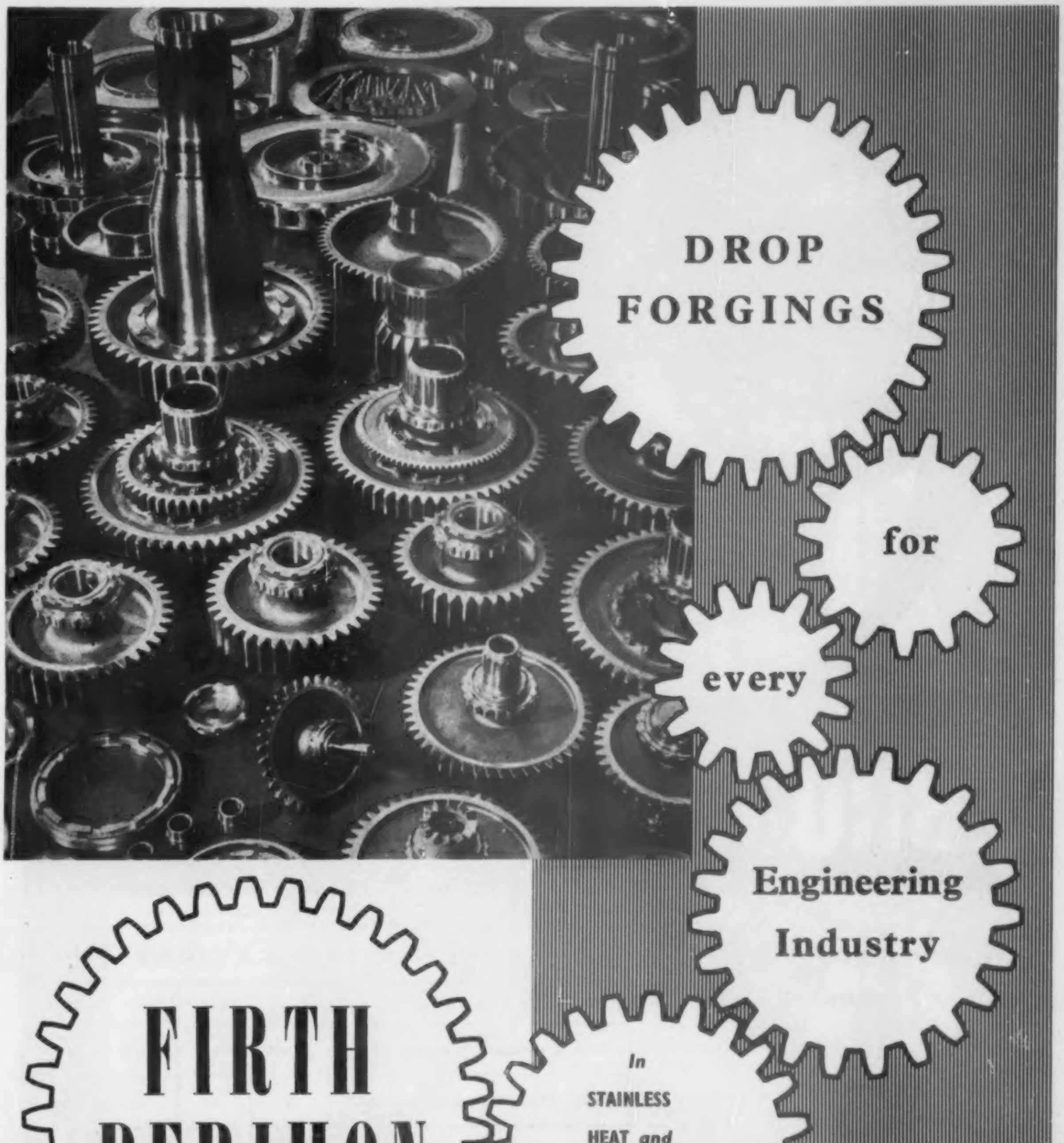
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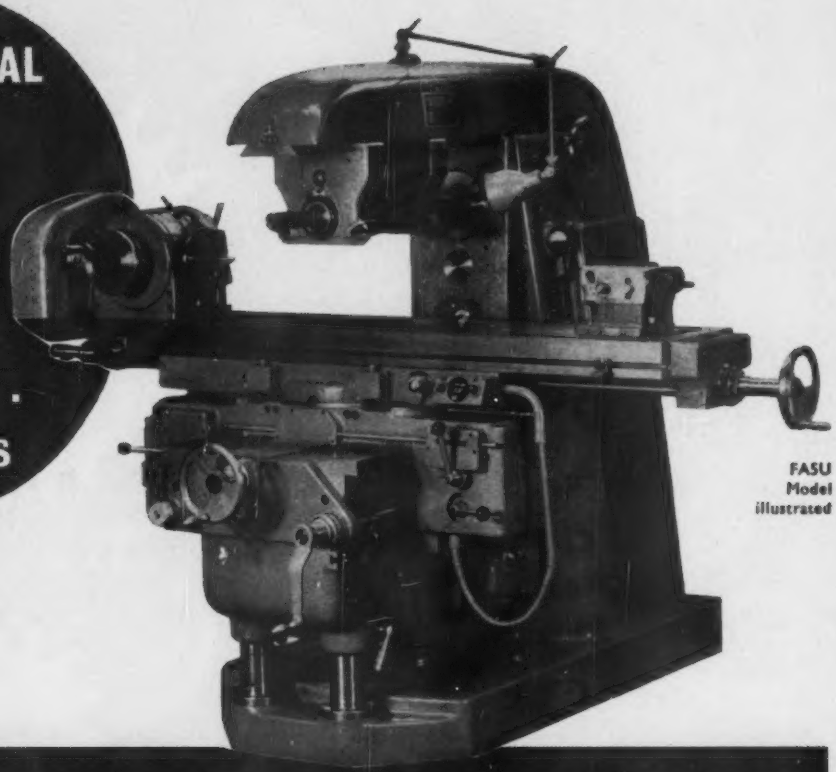

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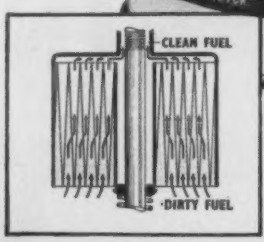

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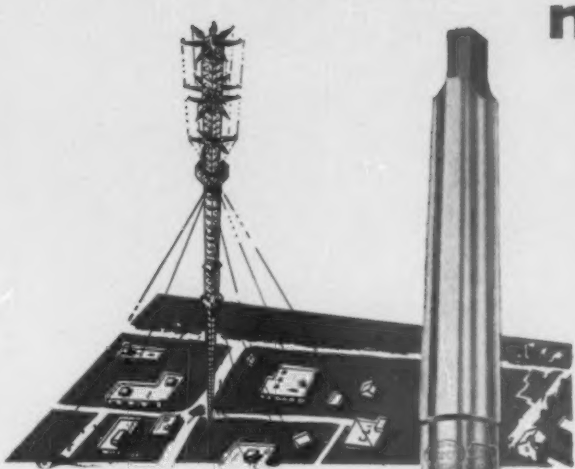

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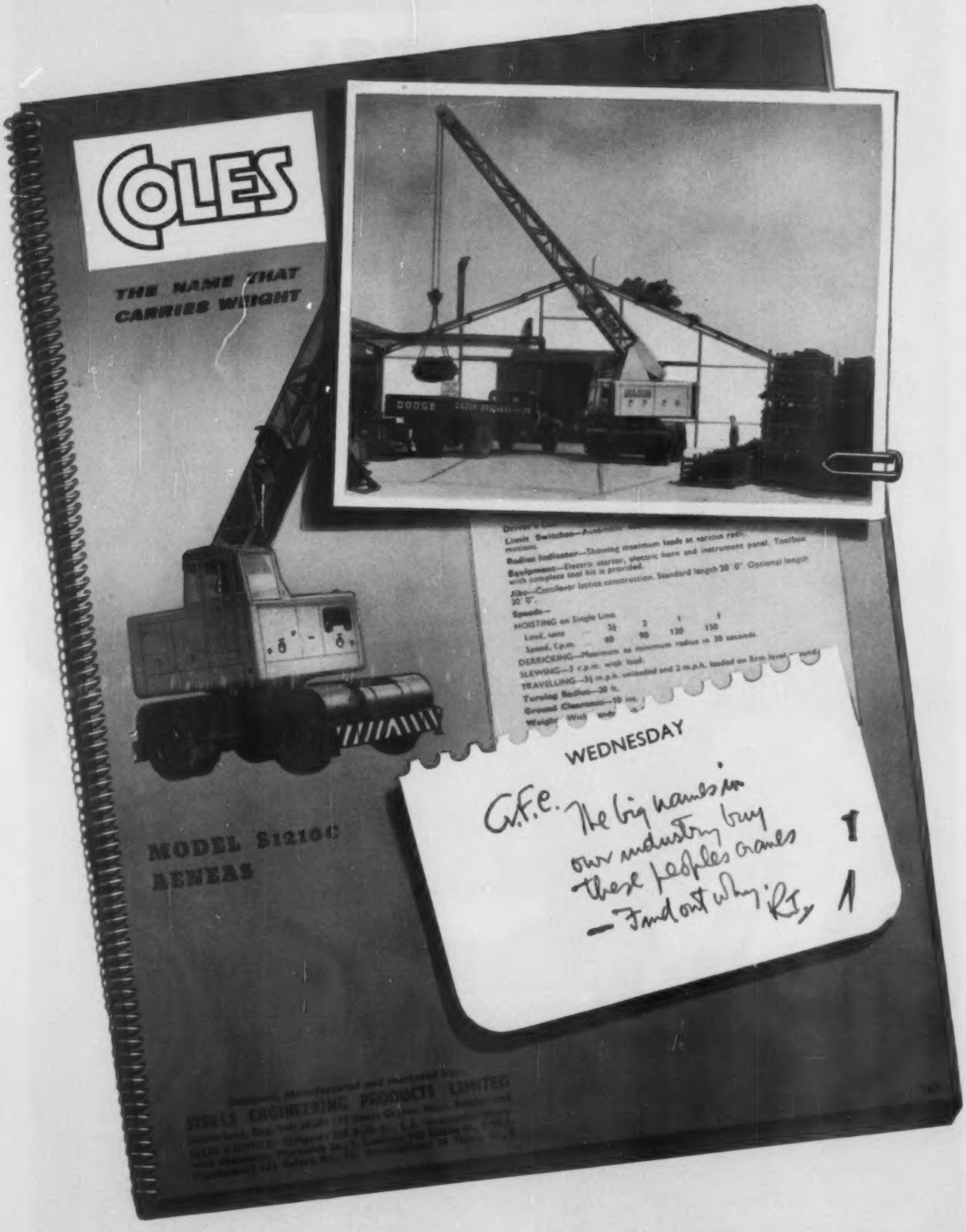
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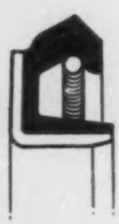

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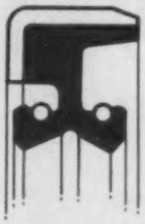

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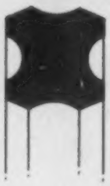

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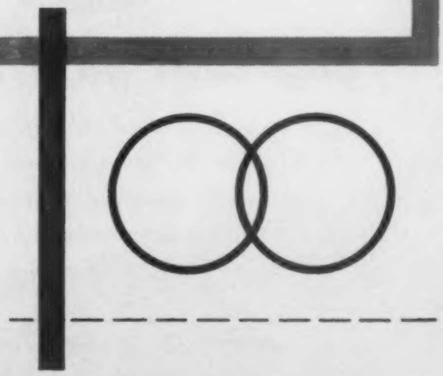

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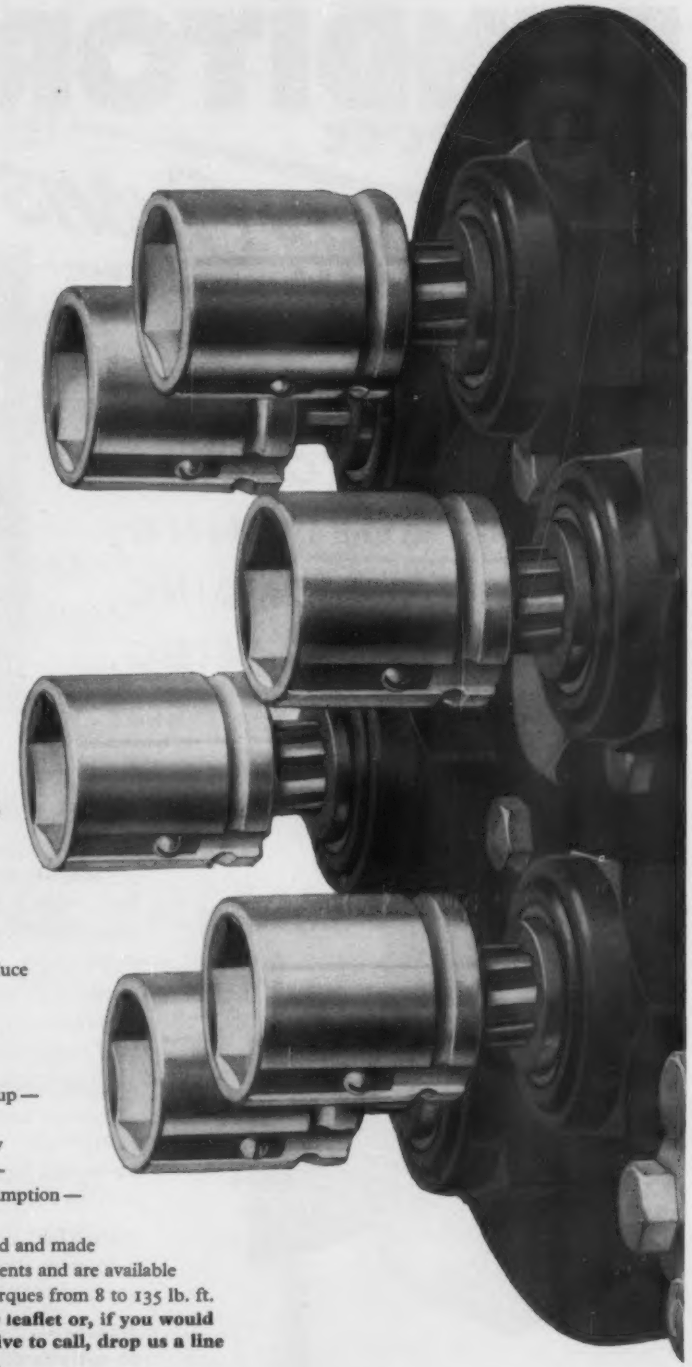

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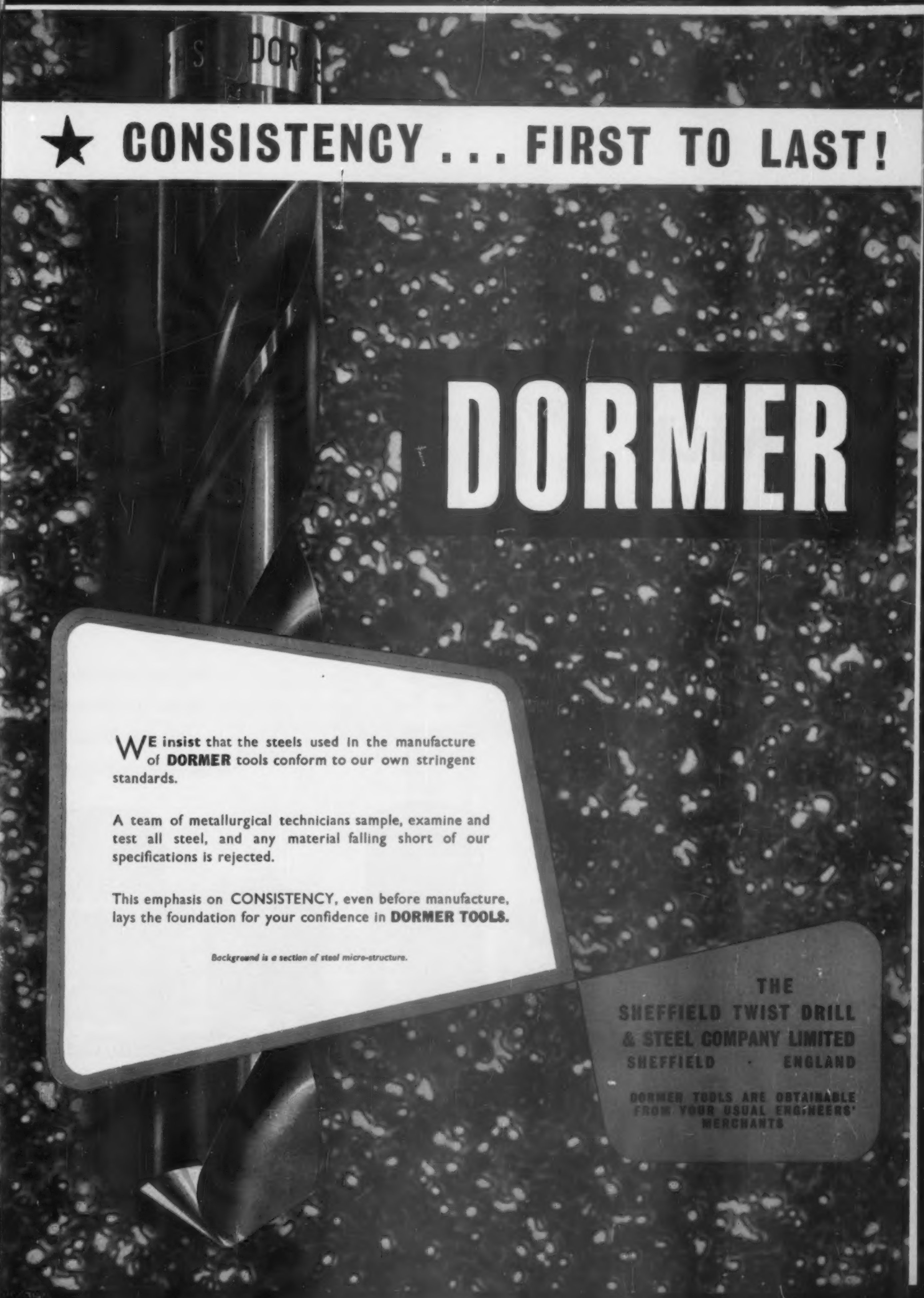
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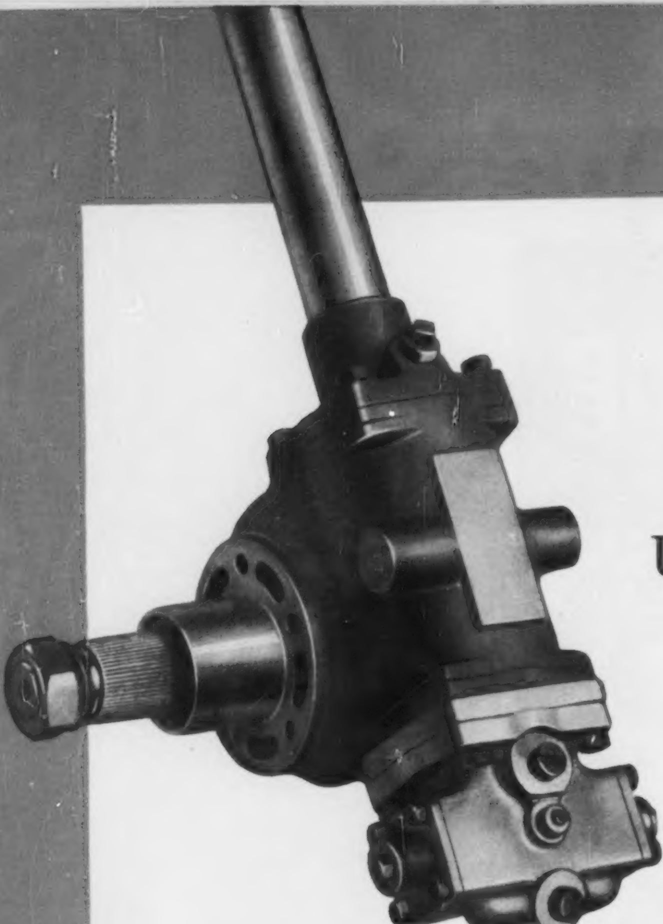

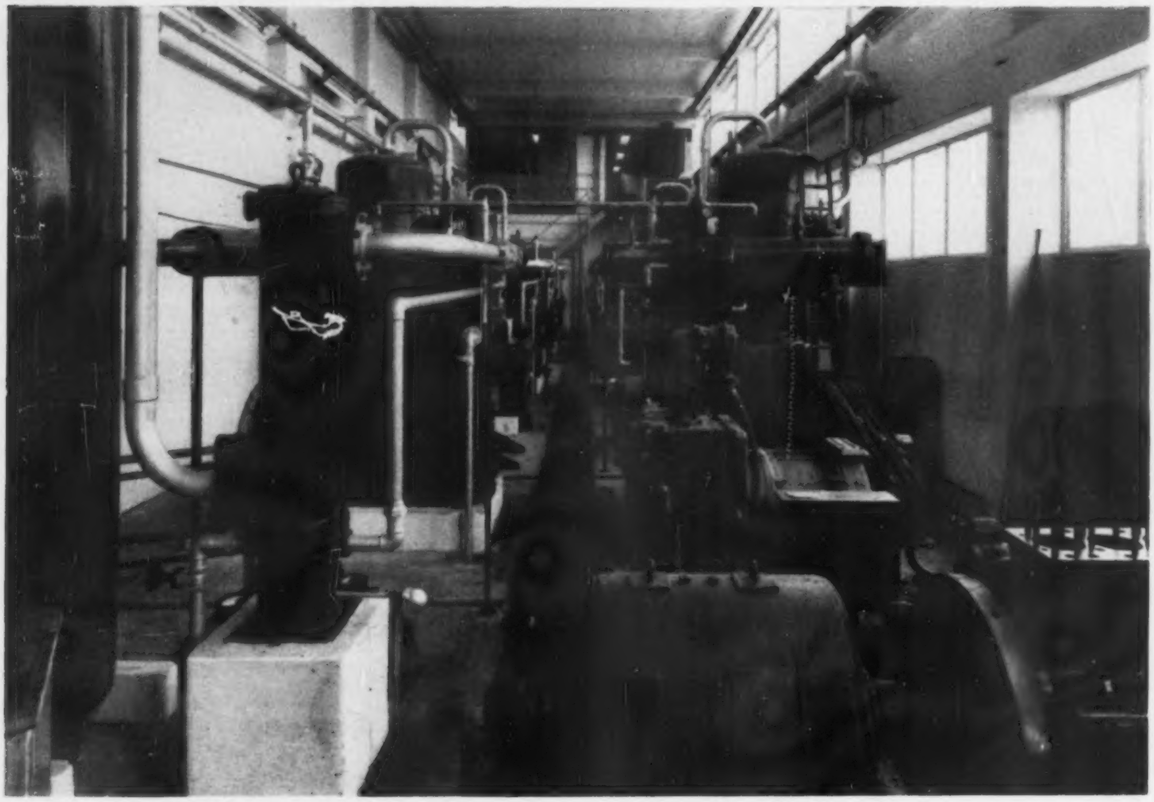

An installation of six "BROOMWADE" type TS Compressors, one with guard removed to show drive, at the Borg-Warner works, Letchworth.

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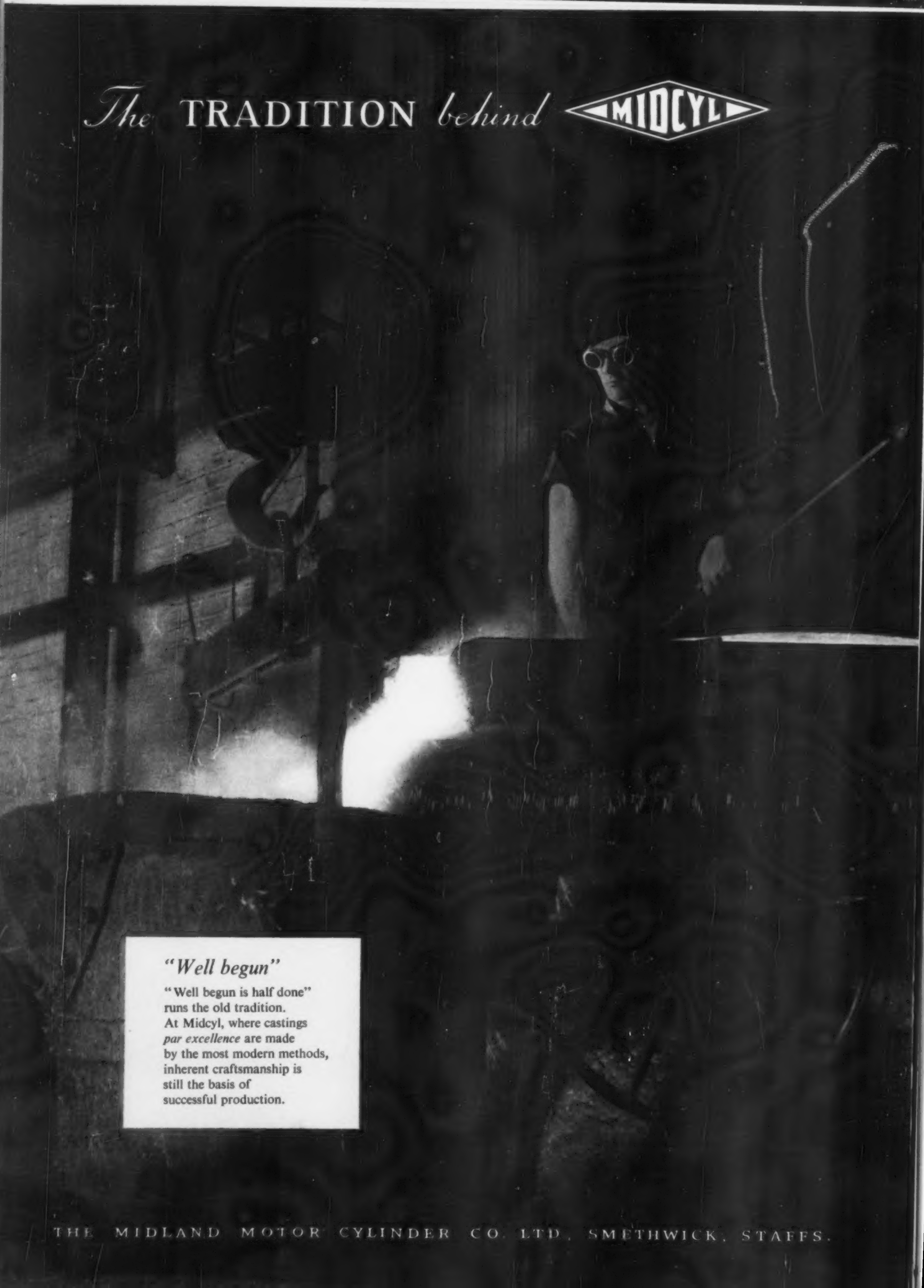
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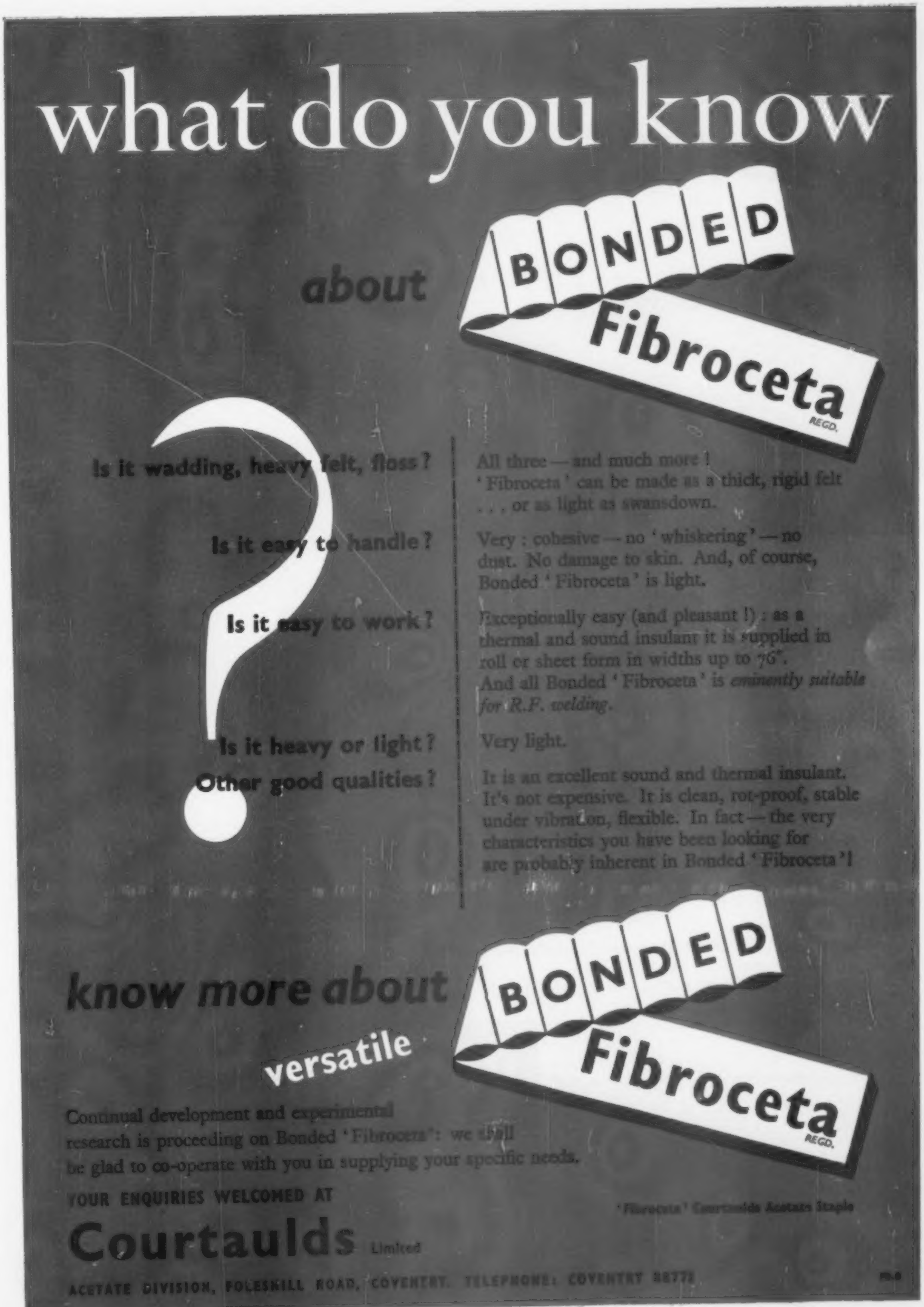
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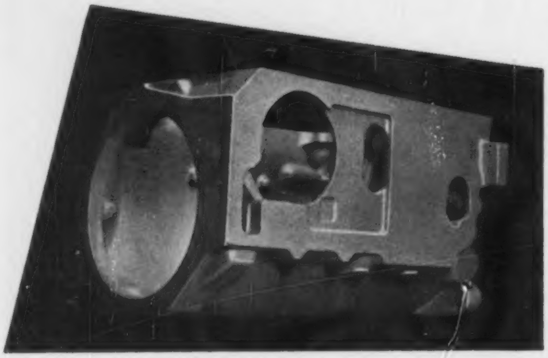

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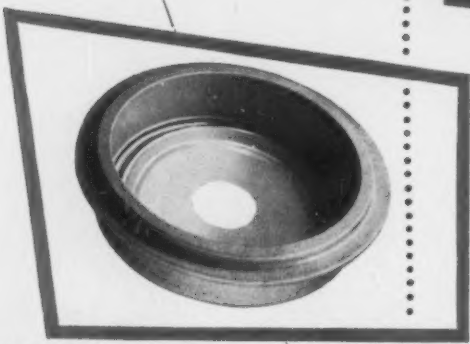

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1

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DESIGN · PRODUCTION · MATERIALS

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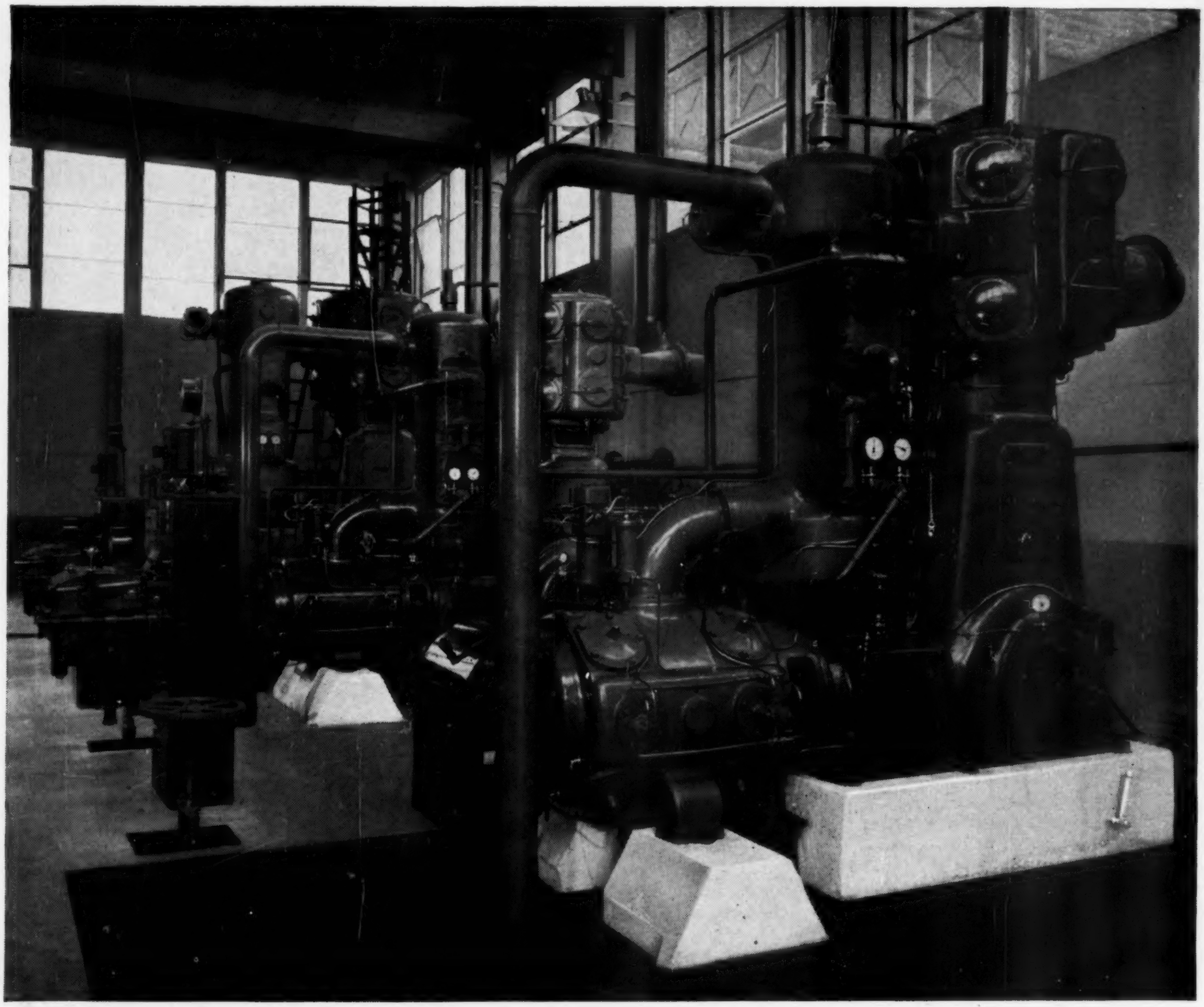

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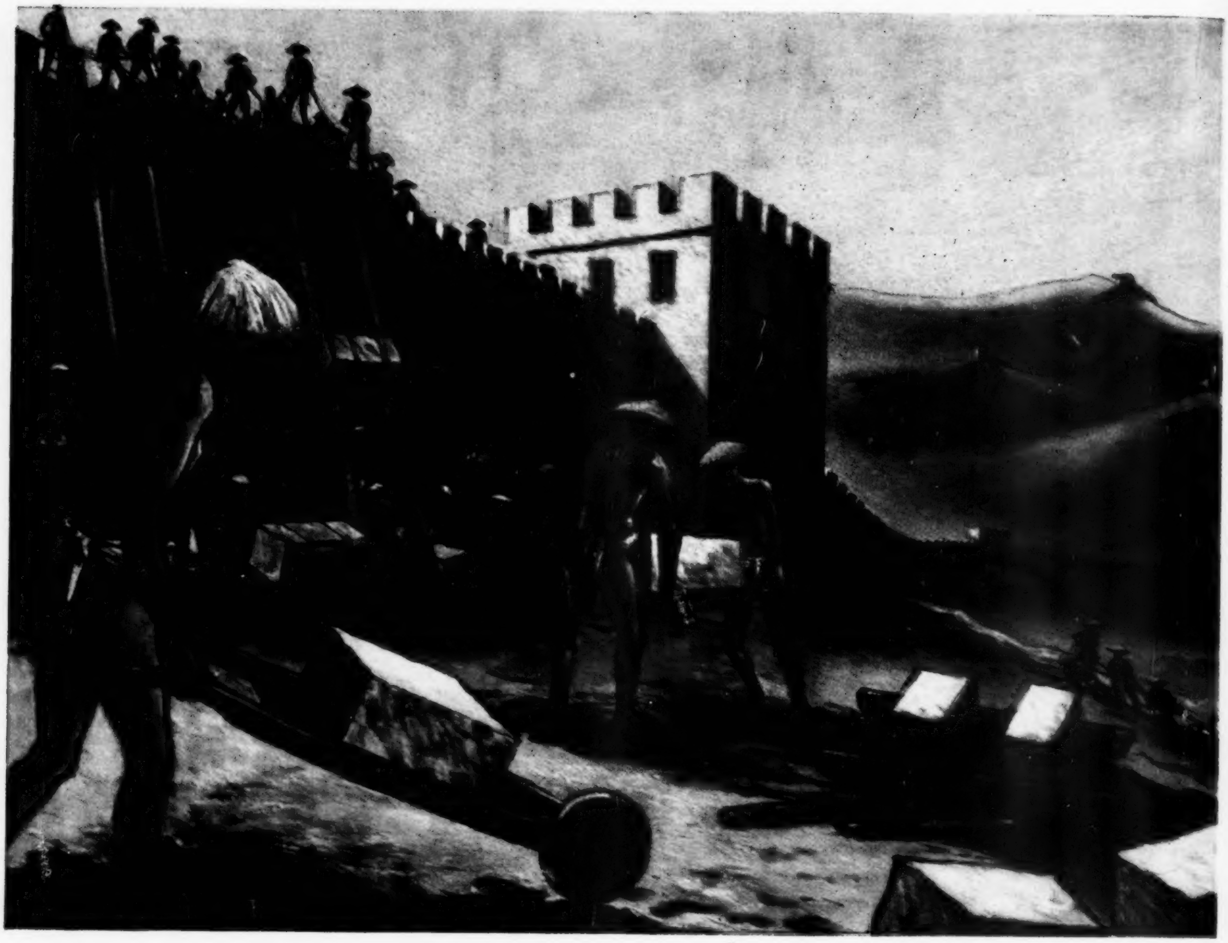

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DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

The Future Outlook

THAT today the outlook is brighter for the British automobile industry than it was at the beginning of 1957 is something for which we can all be thankful. The upsurge in demand that occurred in the last six months of last year has been gratifying and heartening. Even, however, in the early months of 1957 when demand was falling seriously, automobile manufacturers did not lose confidence in the long term. Tremendous expansion plans were continued, although in some cases at a slower tempo, and now the general opinion is that, apart from major political or economic upheavals, the industry can more than hold its own against foreign competition.

These columns are not concerned with political questions, except in so far as they affect the well-being and prosperity of the industry. There is, however, one political matter to which we have referred before and must now refer again. A *sine qua non* for an industry that produces in large quantities for a very competitive market is that there must be a large home market that can take up any excess production when there are recessions in export markets. On this matter it is apposite to quote an independent opinion. In "Britain and Europe," a report prepared by The Economist Intelligence Unit on the probable effects of a European Free Trade Area, it is written that "The cost-reducing effect of an expansion of output may easily swamp increases in raw material or labour costs. This has been the case in France where the level of wages and raw material prices is very high indeed but where efficient productive organization and fuller utilization of capacity have kept costs to a very competitive level. In the long run, costs per unit of output may depend very much more upon the degree of capitalization and the size of the domestic market than upon the price structure of the economy as a whole."

So far as the technical outlook is concerned there is every reason to believe that just as current British road vehicles hold their own against their Continental competitors so will future designs. In general, it is true to say that for several years to come development will consist in modifications of existing design patterns rather than in radical changes.

In the case of power units there is not the slightest evidence that anything will replace the piston engine in the near future, nor for the time being that direct fuel injection will be adopted in place of carburation, except, perhaps, for sports cars and some expensive passenger cars. However, the trend towards higher compression ratios will call for very careful attention to combustion chamber design and breathing characteristics.

There is no doubt that two-pedal control, with either semi-automatic or full-automatic transmission, will soon be the rule rather than the exception. The day of the conventional gearbox with conventional gear-shift is not yet past but it is passing. This was evident at the 1957 Earls Court Motor Show where only one major British manufacturer failed to show at least one model with some form of two-pedal control.

In the case of suspensions, major advances and developments are much more likely for public service and heavy commercial vehicles than for passenger cars. A pointer showing what may be expected was afforded by the 1957 Frankfurt Show, where almost every German manufacturer of heavy vehicles showed at least one model with pneumatic suspension. This type of suspension is particularly suitable for public service vehicles. Its principal advantage is that when used in conjunction with an automatic height regulator, optimum use is made of the space available for wheel deflection regardless of the load carried. In other words, lightly laden or heavily laden the ride is the same. The improved ride makes it attractive for private cars also, but at the moment the need for a compressor is a serious obstruction to its use on passenger cars. Pneumatic suspension, however, offers so many advantages that we cannot afford to let Continental competitors get ahead of us in its use. Fortunately, several British manufacturers have either developed or are developing pneumatic suspension systems.

More efficient engines and improved suspensions will inevitably lead to higher road speeds; improved braking will therefore be necessary. There is, of course, a limit to the size of drum-type brake that can be accommodated within the dimensional limitations imposed by the wheel rims; this may lead to the wider use of disc brakes, particularly on commercial vehicles equipped with air suspension. An increased demand for disc brakes could appreciably reduce the cost-factor advantage now possessed by drum-type brakes.

Cars, smaller than anything they now produce, have been and will likely continue to be the subject of careful study by several major manufacturers. There is considerable divergence of opinion concerning the market for these cars, but in our opinion if such a vehicle is to win wide acceptance in this country, it must be equipped with an engine of not less than 600 cm^3.

Air Suspension on

Products of the Component Suppliers, and a Discussion of Some Features of Their Design

IN the air suspension systems on the vehicles that were described in Part I of this article, the individual components, such as springs and height regulator valves, are supplied by many different manufacturers, most of whom are primarily engaged in the production of tyres and other rubber components. The majority of these firms are associated with companies both in this country and in America, and in all three countries development has been proceeding on parallel lines and there has been a great deal of interchange of information. It follows, therefore, that similar components are in many instances produced in both this country and America.

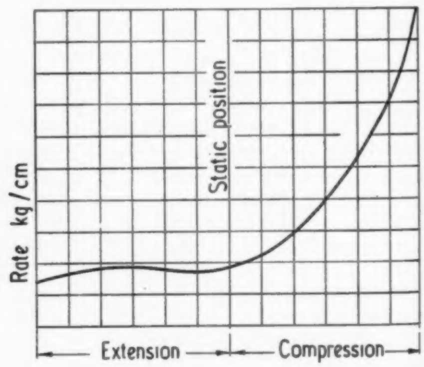
This article is the sequel to one published in the December 1957 issue of Automobile Engineer. *In the first part, the general principles of air suspension and some of the layouts employed on the Continent were discussed. This second part goes on to describe the individual components and deals with some of the problems involved in their design. The article ends with a few comments on two systems that are of particular interest because they require neither a compressor nor any other scurce of air.*

Bosch

Bosch G.m.b.H., of Stuttgart, are developing height regulator valves for air suspension systems. The designs of some of these valve units are not yet finalized and patents are pending, therefore few details are so far available. In general principle, they function in the following manner. Movement of the actuating lever does not immediately actuate the valve, because the motion of the valve is hydraulically damped and there is a flexible element incorporated between the spindle actuated by the lever and the spindle that actuates the valve. An adjustment screw is provided so that the delay period between the movement of the actuating lever and the operation of the valve can be regulated to any period from zero to 20 sec. This adjustment device is simply a conical-ended screw, which projects into and restricts the transfer duct through which the hydraulic fluid passes from one side of the damper piston to the other.

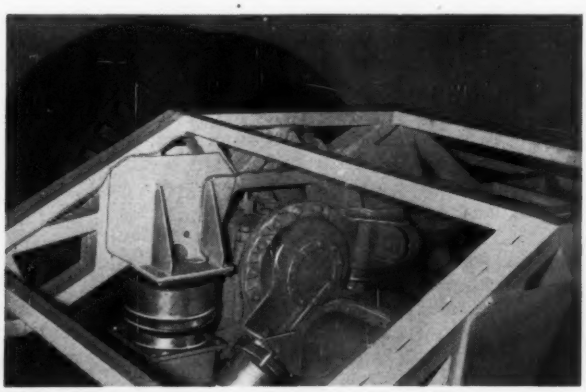
The general arrangement of the valve assembly is not unlike that of the Westinghouse hydraulically damped valve, shown in an accompanying illustration. That is, the air valve and damper piston assembly are mounted one above the other, with their axes parallel, and tongues on them register in slots in a drum, which is rotated by the actuating lever: the axis of this drum is at right angles to the plane containing the axes of the air valves and the damper piston assembly.

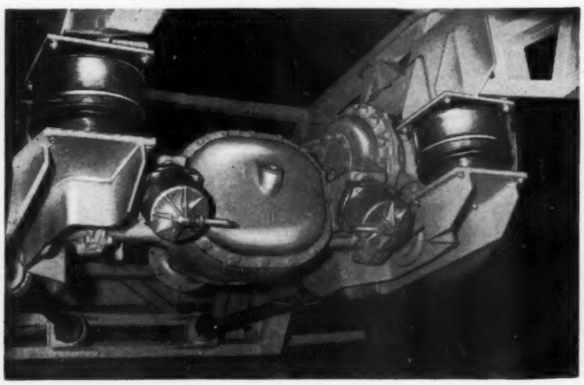
However, the Bosch arrangement differs from the Westinghouse unit in that, as has already been mentioned, the flexible element is interposed between the spindle of the actuating lever and the drum that moves the valve and damper, instead of being incorporated in the actuating linkage. The Bosch element comprises simply three steel balls, which seat between three conical recesses in the end face of the drum and three more in a disc, the disc being spring-loaded against the drum. When the actuating lever is moved, it rotates the retainer disc and, since the valve is damped and therefore cannot immediately move,



Typical characteristic curve of the CEAT Air Spring, showing how a low and relatively constant rate is obtained over a large portion of the working range and the rate rises steeply towards the full bump position

In the two views below, of an air suspension mock-up exhibited on the Viberti stand at the Turin Show, the trailing locating links with spherical rubber end-fittings can be seen. CEAT Air Springs are used and, in conjunction with this double-reduction rear axle, they form a very compact installation



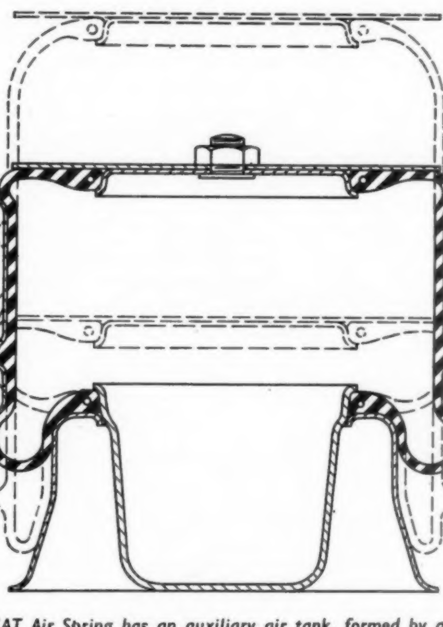


the Continent

and Application to Commercial Vehicles

the conical recesses in the retainer disc move out of alignment with those in the drum, so that the balls ride up their conical surfaces, pushing the retainer plate axially away from the end face of the drum. In this condition, since the spring is tending to force the retainer plate back towards the end face of the drum, the balls are tending to roll down the conical surfaces into their seats. It follows that the seating of the balls can only be effected by rotation of the drum which, in turn can take place only when the damper piston has had time to move the appropriate distance so that the valve can open.

In addition to the valves through which air is supplied



This CEAT Air Spring has an auxiliary air tank, formed by a cupped pressing, in the piston secured to the base of the rubber cylinder

to the spring and released again as required, there are two more in this height regulator valve assembly. One is to prevent the springs from deflating in the event of failure of the pressure supply system. The other is to prevent the air from escaping from the brake system through the air springs in the event of a leakage either from the rubber bellows or from the pipe lines that serve them.

CEAT

In Italy, CEAT-Gomma, of Turin, are producing under licence the Air Spring, which is a registered trade name of the General Tyre and Rubber Co., of America. They state that a wide range of standard sizes is available and that, in addition, special sizes can be designed for particular applications. An unusual feature of the CEAT version of this system is that apparently there is no hydraulic damping to delay the action of the levelling valve. It seems likely, however, that damping is effected by restriction of the air passages in the valve. The manufacturers state that the oscillation of the valve, owing to either vertical displacement of the suspension or rolling of the body, automatically causes air to flow from the reservoir to the spring or to escape from the spring, as appropriate, so that not only is the height of the sprung mass maintained as nearly constant as possible, but adjustments are also made automatically to correct for any tendency to roll during cornering. The disadvantage of the employment of air valves that are not damped hydraulically is, of course, that the air consumption rate tends to be high and the wear and tear on the valves is greater than when damped valves are employed. Undesirable oscillations have been known to occur as a result of undamped air surges caused by repeated valve operation at certain frequencies.

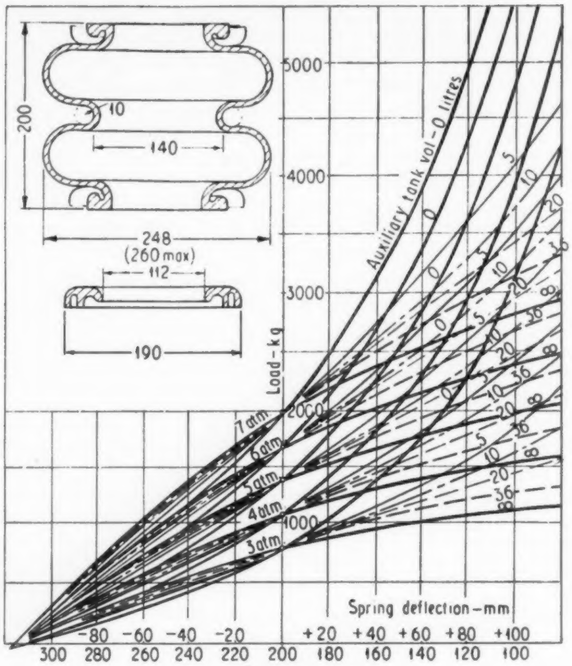
A cross section of the Air Spring is shown in an accompanying illustration. It comprises a cylindrical chamber of rubber reinforced with woven nylon. The cylindrical wall of the chamber is surrounded and reinforced by a steel sleeve, to prevent it from bulging under pressure. There is a hole in each end of the chamber, and the edge of this hole is beaded and reinforced with a steel ring. Clamped to the beading at one end is a circular plate, and to that at the other end is secured a pressed steel piston. A pipe connection, from a hole in the centre of the end plate, is taken to the height regulator valve. The auxiliary reservoir is formed by a cupped steel pressing inserted into a hole in the crown of the piston in such a way that its open end is in communication with the cylindrical chamber.

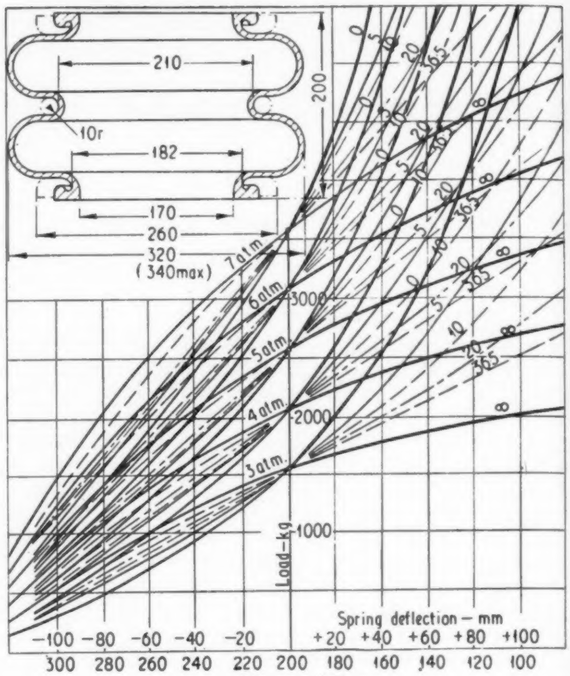
In most installations, the end plate seats under the chassis frame, and the piston is attached to the suspension yoke or link. Therefore, as the wheels rise and fall, the piston moves in and out of the cylinder, the lower end of rubber wall rolling up and down with it. The piston is shaped in such a way that the rate of change of volume of the air in the cylinder varies according to deflection: it can be arranged to suit any particular application. In the illustration showing a typical spring characteristic, it can be seen that the rate increases rapidly as the maximum permissible deflection is approached.

Continental

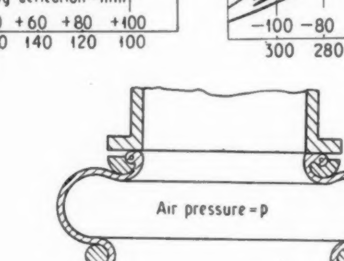
Continental Gummi-Werke A.G. manufacture air springs, of the cylindrical bellows type, the general principles of which have been described in the September 1957 issue of *A.T.Z.* As can be seen from the accompanying illustration, these springs are mostly of the double-convolution type, the two convolutions, or cells, being separated by a circular section, steel ring. This arrangement enables a relatively large range of vertical deflection to be obtained while, at the same time, the diameter of the spring varies little. If the spring is used without an auxiliary air tank, each of its ends is clamped on to its seat by means of a steel ring. However, if an auxiliary tank is used, it generally takes the form of a steel cylindrical extension of the rubber bellows. One end of this cylinder is closed and the other is flanged to form one of the seats for the rubber bellows.

From the illustration of the Continental pneumatic spring, it can be seen that the load P that it can support is dependent on two variables. One is the effective plan area of the seat, multiplied by the pressure, $\pi D^2 p/4$, and the other is the vertical component of the tension in the walls of the bellows, $T \sin \alpha$. The symbols used in these expressions are clearly defined in the illustration. So far as the term $\pi D^2 p/4$ is concerned, it is clear that if the ring clamping the end of the spring on to its seat is of wide cross section, so that the effective area of the seat increases with spring deflection, a rising spring rate is obtained. On the other hand, with a thin ring, this *seating ring form-factor* is virtually constant.

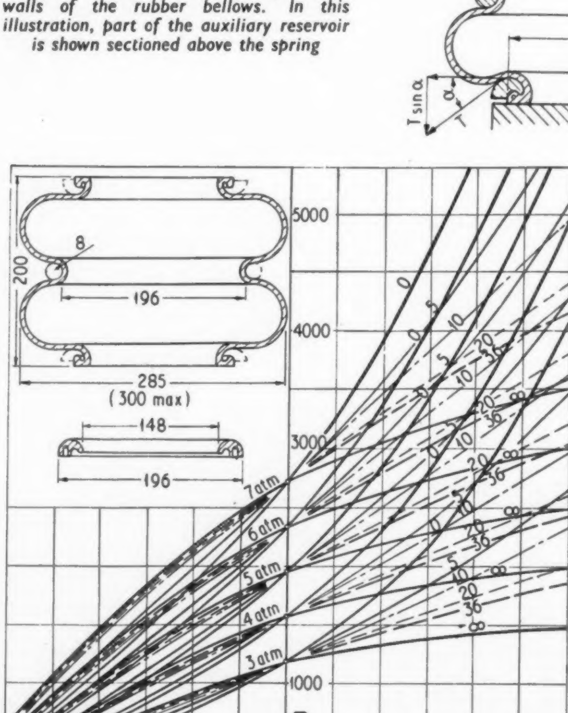


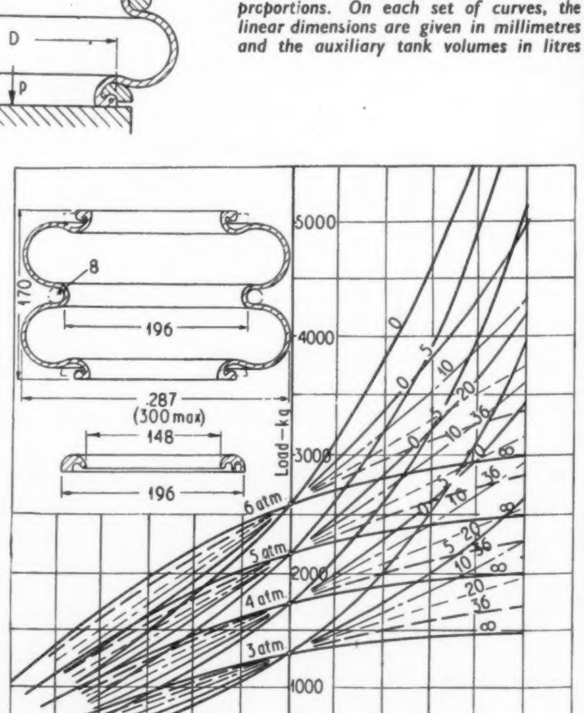


The illustration on the right shows how the rate of the Continental pneumatic spring is dependent upon the air pressure times the plan area of the seat, plus the vertical component of the tension in the cylindrical walls of the rubber bellows. In this illustration, part of the auxiliary reservoir is shown sectioned above the spring



The four sets of curves on this page and the two sets on the opposite page show how the characteristics of the Continental pneumatic springs can be regulated by the employment of bellows and seating rings of different proportions. On each set of curves, the linear dimensions are given in millimetres and the auxiliary tank volumes in litres

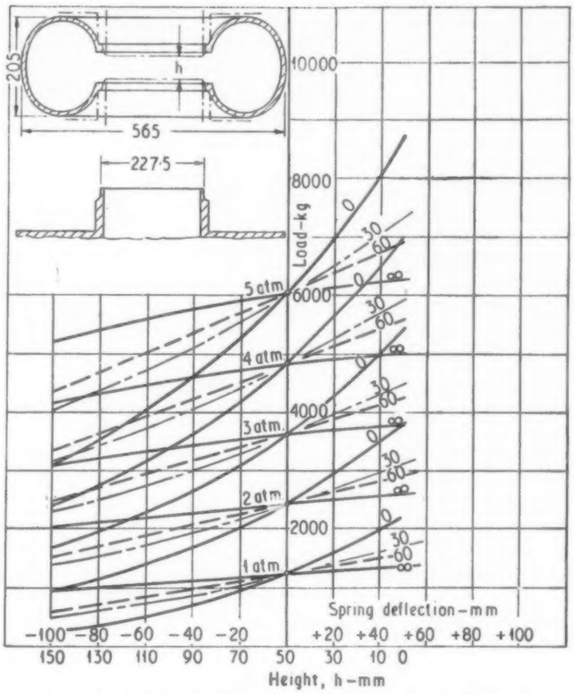


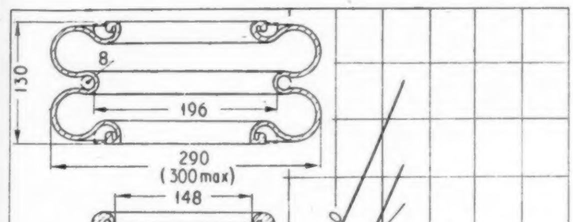


Since α is continually changing, the second term, which might be called the *bellows form-factor*, varies continuously with deflection. The way in which α changes can be regulated to a certain extent by altering the amount by which the periphery of the bellows overhangs that of the seating ring. With a small overhang and a seating ring of wide section, a steeply sloping load deflection curve can be obtained. This slope can be reduced appreciably by increasing the overhang; and a relatively flat curve can be obtained with a large overhang and a ring of narrow section. The width of the ring section affects mostly the lower portion of the load-deflection curve, while the degree of overhang tends to influence almost the whole range, particularly if a narrow section seating ring is employed. A wide section ring is generally employed so that the compression force on the spring will decrease rapidly as the suspension moves to the rebound position.

Yet another factor affects the variation of spring rate, with deflection. This is the fact that, for rapid deflections, the air in the spring is compressed adiabatically, according to the law $pV^n=$ constant, where p is the absolute pressure, V is the volume of air compressed and n is the ratio of the specific heat of air at constant pressure to that at constant volume. It follows that $p_2/p_1=(V_1/V_2)^n$, where the suffices 1 and 2 refer respectively to the conditions before and after deflection. Thus it can be seen that the greater the ratio of $V_1 : V_2$, the more steeply will the load deflection curve rise as deflection increases; in other words, an air spring used without an auxiliary tank will have a more steeply rising characteristic than one with such a tank. Therefore, the slope of the load deflection curve can also be regulated by varying the size of the auxiliary tank.

The effects of all these variables can be seen in the set of six curves, showing the characteristics of springs of different detail design. These curves show that, unless low pressures and therefore unduly large springs are employed, the rate increases too rapidly with deflection when the springs are not used in conjunction with an auxiliary tank. At the other extreme, if an auxiliary tank of infinite volume is employed in conjunction with this type of spring, the load deflection curves are too flat for most applications, particularly at the upper end of the deflection range, where the bellows form-factor is almost a constant, unless impracticably high pressures are adopted. It is of interest to note that when the auxiliary volume is infinity, the variations in the slope of the curves are entirely due to the seating ring and bellows form-factors. In practice, of course, the volume of the auxiliary air tank chosen is the compromise, between the extremes of zero and infinity, which gives the required load deflection characteristics and a convenient pressure.





Dunlop

The Deutsche Dunlop Gummi Compagnie A.G., of Hannau am Main, have done a great deal of development work on air suspension systems. They claim that with their system, the air consumption is extremely small because the height regulator valve operates only to correct for load variations, and it can therefore be employed for passenger cars, even if a compressed air bottle is installed to act as the source of supply for the pneumatic system. In practice, the air pressure in the springs seldom exceeds 5-6 atmospheres, but the highest pressure that the springs are capable of carrying is 7 atmospheres.

Several layouts are envisaged. Among these are systems in which both the front and rear axles are located by V-links and Panhard rods. The apex of the V-link is pivoted on the vehicle frame, while its splayed ends, which are attached to a beam axle, form the seats for the lower ends of two cylindrical, double-convolution bellows pneumatic springs. Two shock absorbers of conventional size are employed on each axle. Cylindrical, auxiliary air tanks form the seats for the upper ends of the springs.

An alternative arrangement for a single-axle rear suspension system is as follows. The conventional leaf springs are replaced by longitudinal beams, the ends of which are swept round in front of and behind the rear wheels, to carry the lower seating pads for the four, cylindrical, double-convolution bellows type, pneumatic springs. This gives a wide spring base, but presumably cross members would be required between the swept ends of these beams. Location of the axle is effected by two trailing links and a V-link. The apex of the V is pivoted on top of the axle casing, while its ends are pivoted on the vehicle frame. One trailing link is

pinned between lugs below each end of the axle; the other ends of these links are pivoted on downward-extending brackets under each side member of the frame. The illustration of this system, in the lower right-hand corner of the opposite page, is of a model shown on the Dunlop stand at the Frankfurt Show.

Knorr-Bremse

Knorr-Bremse G.m.b.H., of Munich, manufacture a height regulator valve for air springs. As can be seen from the accompanying diagram and the detailed illustration of the unit, movement of the actuating arm rotates a cam, which applies a torque to the valve operating rocker. However, since this rocker is connected at one end to the piston of a hydraulic damper, it cannot move immediately. Instead, the cam compresses a pair of spring-loaded plungers housed in the centre of the rocker. Continued application of the torque, through the spring-loaded plungers to the rocker, causes the piston of the damper to move slowly in its cylinder, displacing oil through a transfer port from one side of the piston to the other. This allows the valve actuating rocker to rotate about the cam spindle, on which it is mounted, until the plungers are fully extended again. In this condition, the rocker bears on the end of one or other of the two valve stems, according to the direction of the applied torque. One of these valves regulates the entry of compressed air to the spring, while the other controls a port to atmosphere. The transfer passage from one side of the damper piston to the other has a conical-ended metering screw in it. Adjustment of this screw determines the delay period before the appropriate valve is opened. In the illustration of a mock-up of a swing axle suspension system, produced for display purposes, the Knorr-Bremse height regulator valve is shown in the foreground to the left of the spring. The radius rod, which guides the swing axle, is just visible behind the spring.

Metzeler

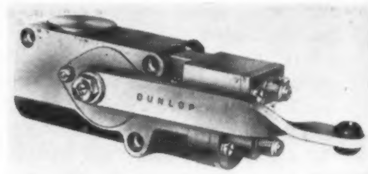
The Metzeler Gummiwerke A.G., of Munich, manufacture a spring similar to the CEAT unit already described. In the accompanying diagrammatic illustration, it is shown installed in a typical motor car front suspension system. The spring comprises a closed cylinder of rubber reinforced with cord plies, like a tyre, and a pressed steel piston of conical form, which moves in and out of its lower end, rolling the lower end wall of the cylinder with it, as the wheel rises and falls. A steel sleeve reinforces the periphery of the cylinder. The conical form of the piston gives a spring rate that progressively increases with deflection of the assembly.

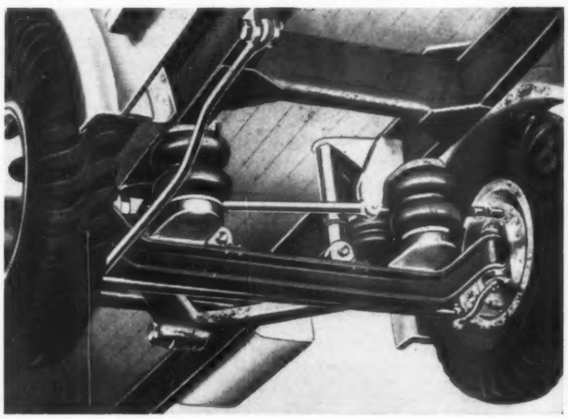
The advantages claimed for this spring are that it enables a larger deflection to be obtained than is practicable with either circular bellows or elongated bags; it is compact; almost instantaneous operation is obtained, and it is said to have a long life. As can be seen from the illustration in the top right-hand corner of the opposite page, this type of spring is well suited for application to a transverse wishbone type front suspension. In fact, it might even be practical to substitute it for the coil spring in an existing suspension system, provided the diameter of the pneumatic spring is not too great. This spring is, of course, equally well suited to other suspension arrangements.

Phoenix

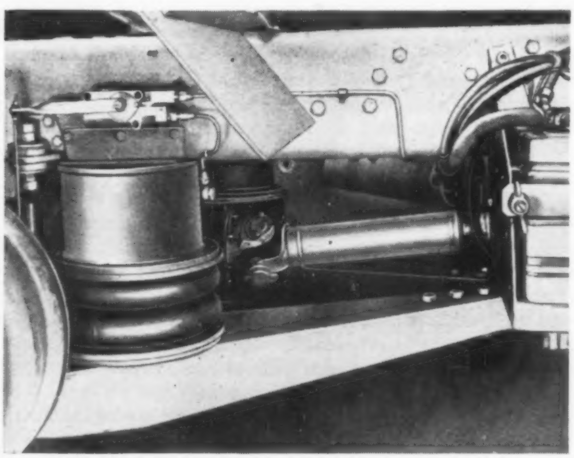
The Phoenix Gummiwerke, of Hamburg-Harburg, are producing under licence the Firestone air suspension systems. Development of air suspension has been carried out for many years by The Firestone and Rubber Co.: for example, a paper entitled "Air Springs—Tomorrow's Ride," by Roy W. Brown, of this Company was published in the 1936 issue of the *Journal of the Society of Automobile Engineers.*

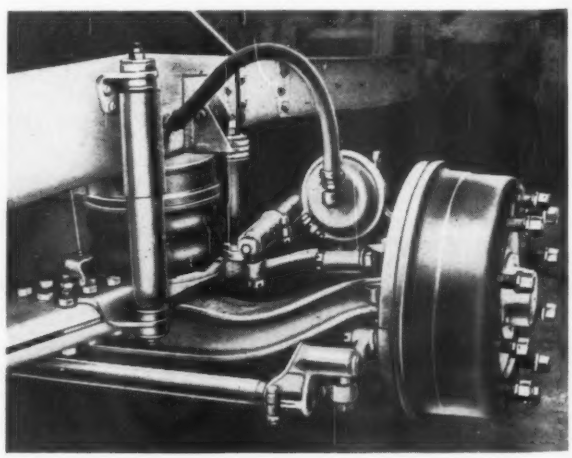
Dunlop manufacture a damped, automatic levelling valve for use in conjunction with pneumatic suspension

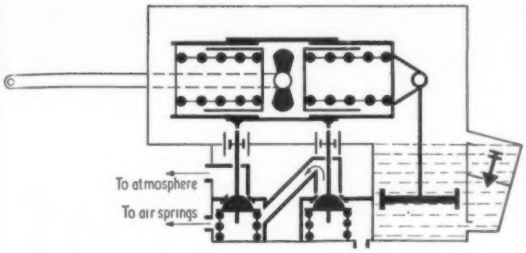


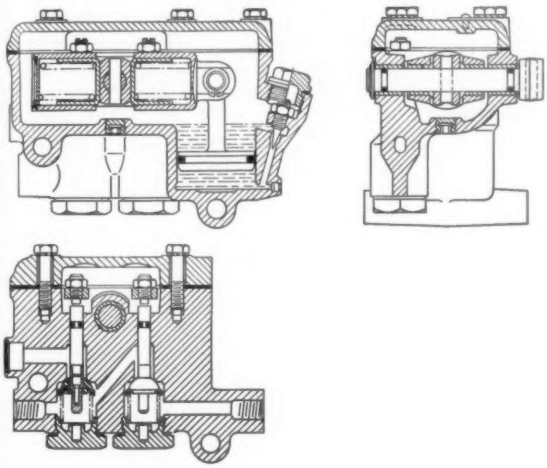


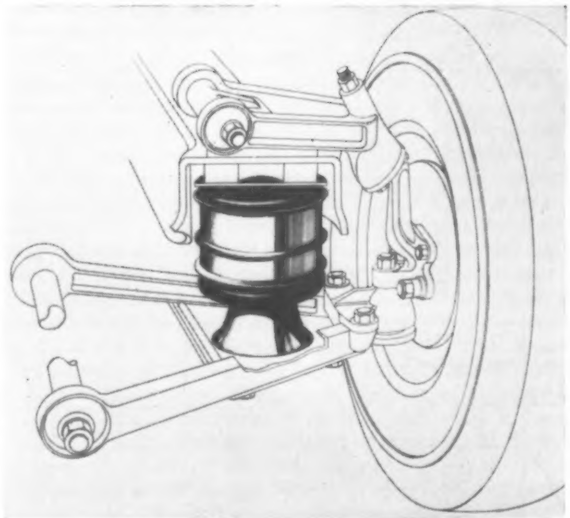
These three illustrations, one on the left and two on the right, show pneumatic suspension layouts proposed by Dunlop. That on the left is, of course, of a rear suspension arrangement, while the other two are for a front axle, a similar method of location being employed in each case











Above: *Diagrammatic illustration showing the Metzeler pneumatic spring applied to a wishbone type front suspension arrangement*

Top, left: *A diagram showing the principle of operation of the Knorr-Bremse, hydraulically-damped, automatic levelling valve*

Bottom, left: *Layout of the Knorr-Bremse height regulator valve. Spring-load plungers are interposed between the cam and rocker*

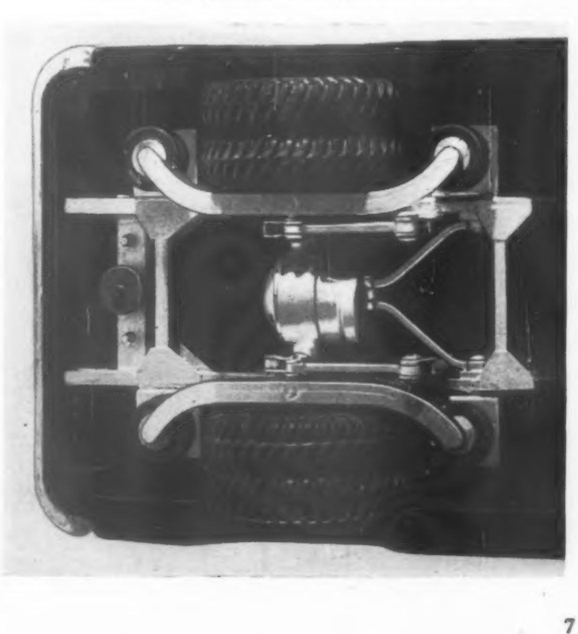
An interesting detail described in that paper is the pendulum valve, which has been mentioned again recently in technical articles on air suspension published on the Continent. It is normally inserted between the pneumatic spring and the auxiliary tank. As can be seen from the accompanying illustration, it is a simple disc type valve, the disc being free to move vertically between two seats. When it moves on to the upper seat, it completely closes the passage between the spring and the reservoir: in other words, it acts as a non-return valve during upward movements of the suspension. Similarly, when it moves on to the lower seat, it again closes the main passage, but in this case, auxiliary passages formed by holes drilled round the periphery of the seat remain open. In these circumstances, that is on the rebound stroke of the suspension, the flow of air from the auxiliary tank to the main tank is restricted, so that the rate of pressure drop for a given rebound movement of the suspension during a short period of time is high.

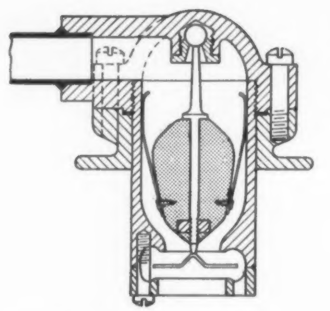
Normally, when the suspension is deflected upwards, a pendulum device prevents the valve from rising above its mid-position, thus holding it clear of the upper seat. In this mid-position of the valve, the flow of air between the spring and the auxiliary tank is least restricted. It follows that the spring rate during the bump stroke is lower than during the rebound stroke. When the vehicle is cornering, however, the pendulum, under the action of centrifugal force, swings to one side and allows the valve to come on to its upper seat during bump strokes. Thus, by isolating the spring from the auxiliary tank, it increases the spring rate and therefore improves the roll resistance. The valve can also be designed so that the pendulum releases the valve during acceleration or braking. This increases longitudinal stability; for example, it can be arranged to prevent nosing-down during braking.

Despite the obvious advantages of this valve, it also has a number of disadvantages, which perhaps account for the fact that it has not so far been widely used. Nevertheless, there is probably a case for re-examining its potentialities in the light of the extensive development work currently in progress in connection with air suspension systems. The first point to be considered is whether the incorporation of such a valve is, in fact, desirable: although it increases the roll stiffness when the non-return valve is closed, it at the same time increases the natural frequency and might possibily tend to give an uncomfortable ride. No doubt the answer to this question has been provided by development work that has already been carried out.

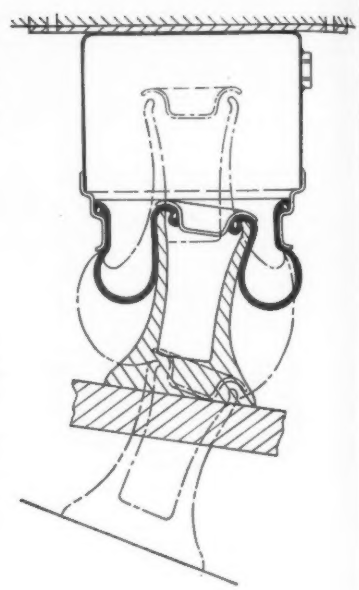
With a valve of this type incorporated in the passage between the air bellows and the reservoir, it would appear to be difficult to obtain an entirely free flow of air between the two components, should this be desired. It is possible, however, that the degree of obstruction might be limited sufficiently to avoid affecting adversely the suspension characteristics. Probably the key question, the answer to

Illustration of a display model, showing the general layout of a Dunlop rear suspension system for a conventional driven axle

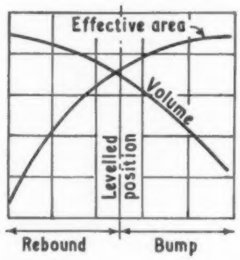


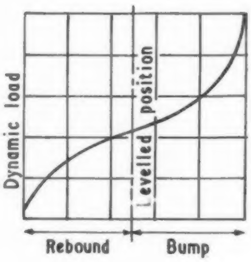


Firestone pendulum valve, which, under the influence of centrifugal force, allows a non-return valve to close the port between the pneumatic spring and auxiliary reservoir



A cast piston is shown in this illustration of the Firestone Airide spring. However, in practice, it is probable that a pressed steel piston would be used

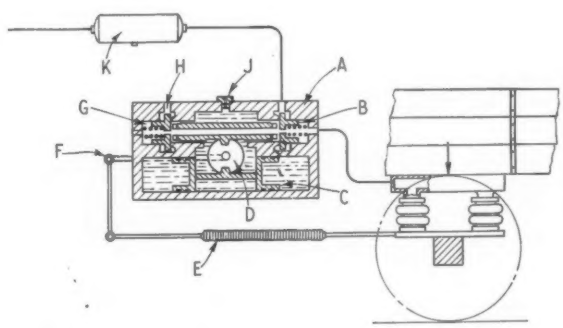




Left: Effective area and volume characteristics, and the dynamic load curve of a typical Firestone spring

Below: The Westinghouse hydraulically-damped valve

A levelling valve assembly; **B** inlet valve; **C** damper piston assembly; **D** actuating drum; **E** spring element; **F** actuating arm; **G** exhaust valve; **H** exhaust port to atmosphere; **J** filler plug; **K** air reservoir

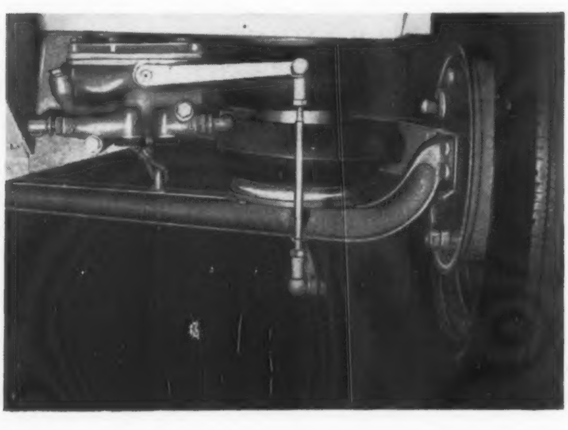


which decides if such a valve is to be used, is whether adequate roll stiffness can be obtained simply by the use of a single-convolution type bellows or a cylinder, piston and seal type spring arranged in such a way that their rates increase with deflection.

It should be noted that the effect of employing this type of valve differs from that obtained by the incorporation of an anti-roll bar. With an anti-roll bar, the effective spring rate is increased not only during cornering, but also when one wheel is deflected alone, instead of both simultaneously. On the other hand, the non-return valve comes into operation only during cornering and therefore the suspension rate remains appropriately low for all vertical deflections of the wheels when the vehicle is travelling in a straight line. Thus, the ride obtained with a system incorporating a non-return valve should be better than that with a suspension in which roll stiffness is obtained by the incorporation of an anti-roll bar.

The latest type of Firestone Airide spring and its typical performance curves are shown in accompanying illustrations. This spring is of the single-convolution type, or perhaps it might better be described as a cylinder and piston with a rolling diaphragm seal. The actual form of the cylinder and piston varies according to the application: for example, the piston can be a pressing, and the cylinder can be formed by appropriately pressed parts of the vehicle structure. The piston shown in the illustration is of a shape designed to give a rate that increases rapidly towards the upper limit of the bump stroke and decreases rapidly towards the lower limit of the rebound stroke. Among the advantages of this type of spring, as compared with the double-convolution type, are that it is easier to install, it can be designed to give lower frequencies and better characteristics, and its overall height is relatively low so installation presents fewer difficulties.

A mock-up showing the Metzeler, hydraulically-damped, automatic height regulator valve, installed in conjunction with a pneumatic spring and a swing axle type front suspension arrangement



Westinghouse

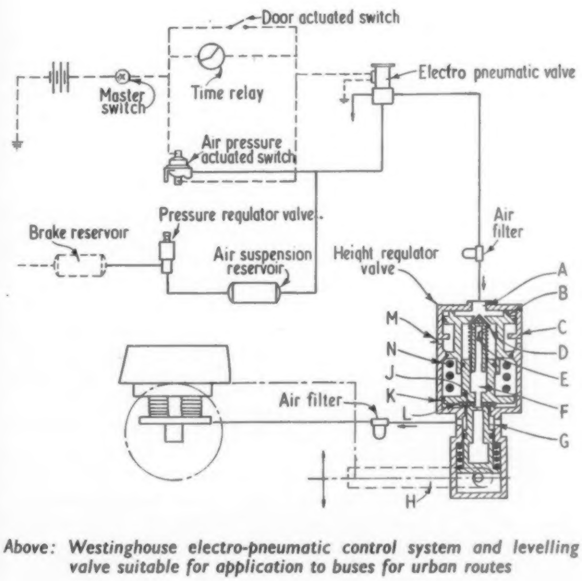
An interesting feature of the Westinghouse air suspension systems is that they can be employed in conjunction with a valve that automatically regulates the degree of braking obtained on the road wheels, according to the load carried. This valve is intended primarily for use with trailers and it has the following advantages. It enables the best possible relationship to be maintained between the braking force and the vehicle load, so that the stopping distance is minimized, while at the same time there is less risk of the wheels locking and the vehicle and trailer jack-knifing. The adjustment is made automatically, so there is no danger of accidents being caused, as there is with hand-operated systems, because the driver forgets to make the adjustment. Tyre wear is minimized because of the absence of wheel locking. It is also possible, by using these valves, to regulate the distribution of braking effort between the front and rear wheels, so that the maximum possible braking efficiency is obtained in reverse as well as in the forward direction. In other words, the arrangement, whereby perhaps 60 per cent

of the braking effort is effected at the front wheels and only 40 per cent on the rear, can be automatically reversed when the brakes stop the vehicle from running backwards.

The Westinghouse, hydraulically, damped, automatic height regulator valve for use in conjunction with pneumatic springs is shown diagrammatically in an accompanying illustration. Its actuating arm is connected to a drop link which, in turn, is connected to a flexible arm fixed to the unsprung portion of the suspension assembly. This flexible arm is formed by a close-wound coil spring.

In the body of the valve assembly, the valve-stem and double-acting damper piston assembly are horizontal, one directly above the other, with their axes parallel. Between the two is the actuating drum, which is mounted on a spindle rotated by the actuating arm. The axis about which this drum rotates is normal to the plane containing the axes of the valve stem and damper piston assembly. A tongue projecting radially from the valve-stem and another projecting from the damper piston assembly register in diametrically-opposite slots in the actuating drum: thus, the damper piston assembly and the valve-stem move simultaneously in opposite directions. An advantage of this horizontal valve and damper layout is that the actions of neither the valve nor the damper are affected by vertical accelerations experienced when the vehicle is operated over a rough terrain.

There are two valves housed in chambers one at each end of the hollow valve-stem. These valves are retained on their seats by compression springs interposed between them and the ends of the valve body. As can be seen from the diagram, connections are taken from the pneumatic spring and the air reservoir into the chamber at one end of the body, while the chamber at the other end has only an outlet port to atmosphere. When the load on the pneumatic spring is increased, the unsprung portion of the suspension assembly deflects upwards, relative to the sprung portion, and the link E moves with it. However, since this link is a close-wound coil spring, and the motion of the actuating arm and drop link is damped, it bends initially and then gradually straightens as the damper allows the valve stem to move to the right, towards the inlet valve at that end of the unit.



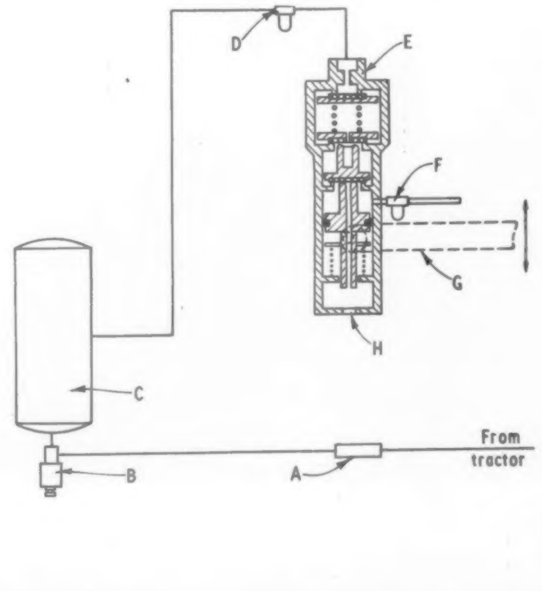
Above: Westinghouse electro-pneumatic control system and levelling valve suitable for application to buses for urban routes

After the delay period, the end of the stem comes up against the inlet valve and lifts it off its seat. This allows air from the reservoir to pass into an annular chamber round the valve-stem and thence through radial holes into the stem. From there, the air passes through an axial hole in the centre of the inlet valve into the chamber to the right, and thence into the pneumatic spring. When the spring has extended sufficiently to restore the suspension to its normal position, the valve-stem is centralized and the valve is again seated.

If the load in the vehicle is decreased, so that the spring height becomes too great, the valve-stem is moved in a similar manner, but to the left. This unseats the exhaust valve, which is that on the left-hand side, and allows air to pass out from the pneumatic spring, through the hole in the centre of the right-hand valve, along the bore of the hollow valve-steam, out through the radial holes at the left-hand end of this stem and thence through the port to atmosphere. The axial hole in the centre of the left-hand valve is incorporated to ensure that the pressures on each side of it are at all times balanced. If the hole were not there, this valve would be blown off its seat when the stem is moved to the right and the valve at the other end unseated.

Some economy of air consumption is obtained with the Westinghouse electro-pneumatic control for regulating air suspension systems on buses, as shown in an accompanying illustration, despite the fact that it is not hydraulically damped. The operation of this Westinghouse control is based on the fact that the load on the vehicle can only be varied when the doors are open.

An electro-pneumatic valve has been interposed between the air reservoir and the automatic height regulator valve. There are three electric circuits, all in parallel, to actuate this electro-pneumatic valve. One is made only when the doors are opened. The second is a time-relay, which causes the valve to open for 10 seconds every 3 minutes, so that compensation can be made for the changes in temperature and for leakage from the pneumatic system. Finally, the third circuit is through a switch actuated by a diaphragm subject to the pressure in the air supply system. When this pressure is less than 4 atmospheres, the circuit is closed, so that air can pass continuously from the compressor, through the reservoir and the height regulator valve, into the pneumatic springs. This circuit is incorporated so that if the pressure in the reservoir is too low when the vehicle leaves the depot at the beginning of its run the air is supplied continuously from the compressor, through the brake reservoir and then the air suspension reservoir, to build up the pressure in the pneumatic springs; otherwise the build-up could take place only during the periods of 10 sec when the valve is opened by the time-relay section of the circuit.



A non-return valve; **B** pressure regulator valve; **C** trailer air reservoir; **D** air filter; **E** height regulator valve; **F** air filter; **G** actuating lever

Right: A system, recommended by Westinghouse, for trailer suspension

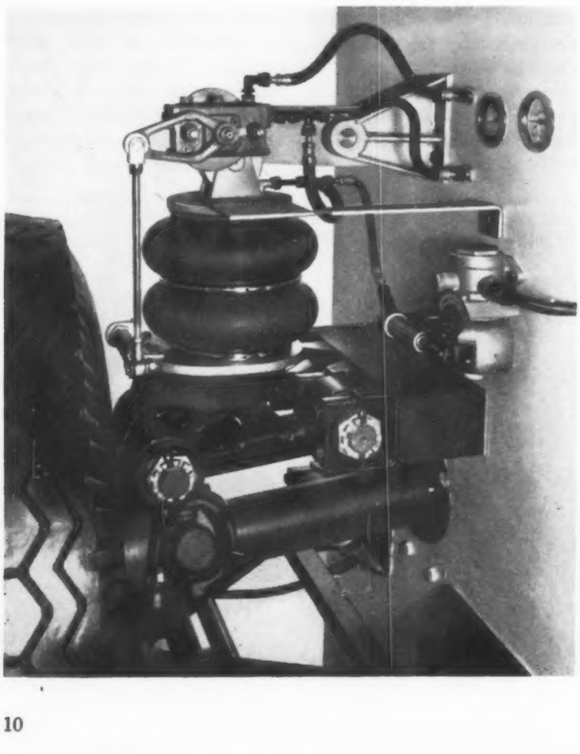
The arrangement of the pneumatic circuit recommended by Westinghouse is as follows. Air is taken from the reservoir for the brake system, through a pressure reduction valve, set to pass the air on at a pressure of about 4·2 atmospheres to the reservoir for the pneumatic suspension. The outlet from this reservoir goes to a T-junction, one arm of which is connected to the pressure-actuated switch of the electrical circuit and the other to the electro-pneumatic valve. From the electro-pneumatic valve, the air passes through a filter to the height regulator valve and thence through another filter to the pneumatic springs.

It appears that damping is effected solely by restriction of the air passages in the height regulator valve. The method of operation of this valve can be seen from the accompanying diagram. Air enters the chamber A, forcing the piston B down on to its stop C. At the same time, the air can go past the non-return valve D, through the small holes E into the chamber F. If the position of the actuating lever H is such that valve G is clear of the seat J, the air goes from the chamber F, past the valve G into the pipeline to the pneumatic springs. On the other hand, if the position of the lever is such that the valve G has been lifted, the air in chamber F can go no further; in fact, the valve K is lifted off its seat L, so that air can be exhausted from the pneumatic springs, past the periphery of the valve K and the lower flange of the piston B, and through the exhaust ports to atmosphere.

In the diagram, the piston B is shown in its uppermost position, in which its lower flange seals off the passage to atmosphere. It is held in this position by the spring N when the pressure of the air supplied from the reservoir is low. In these circumstances, the compression in the spring that retains the non-return valve on its seat is light so that it offers a minimum of opposition to the passage of air into the chamber F.

Westinghouse manufacture a different type of height regulator valve for trailers. As can be seen from the accompanying illustration, air is taken from the pneumatic system on the tractor, through a non-return valve mounted on the trailer, to a pressure regulator valve, and thence into a reservoir. The outlet from the reservoir goes to an air filter and then through the height regulator valve and another air filter to the pneumatic suspension springs of the towed vehicle.

A mock-up of the Westinghouse air suspension system, with a trailer brake pressure regulator valve on the right. Where it is practicable to mount the pneumatic spring directly over the king pin, the loads on the suspension links and their bearings are greatly reduced



In the height regulator valve, the air first enters a cylindrical chamber that has two ports, one at its upper and one at its lower end. These ports are closed by two plate-valves with a compression spring interposed between them to force them on to their seats. The upper of these two valves, which seats on the inlet port, acts simply as a non-return valve, while the lower one is actuated by the upward-projecting stem of a third valve, housed in a chamber below it. This third valve, in turn, is actuated by a hollow push rod, which is moved up and down by the lever connected to the unsprung portion of the suspension assembly. When this push rod is lifted, its upper end seats on the lower of the three valves, which it then lifts, pushing the intermediate valve up off its seat. This allows air to flow from the upper chamber into the lower one, and thence through an outlet to the pneumatic springs. As the spring extends, the actuating lever lowers the hollow push rod until the inlet valves are closed. Further downward movement of this hollow rod lowers its upper end from its seat and allows air to flow back from the pneumatic springs, through its bore to atmosphere.

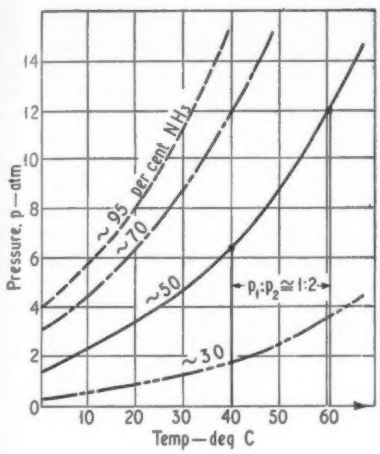
Miscellaneous

Some interesting development work has been carried out with rail cars on which refrigerant gas, such as ammonia, is used instead of air, in bellows type springs. This system, which has been described in the September 1956 issue of *Eisenbahn Technische Rundschau*, has a number of advantages, as compared with the more commonly employed systems, in which air is used as the springing medium. First, it is a closed system, the spring height being regulated by varying the temperature of the refrigerant vapour, and therefore a compressor is not needed. It would appear that the temperature can be controlled either by employing an electrical heating device or by using a heat exchanger, through which exhaust gas from the engine is passed. With the latter arrangement, the heat exchanger would have to be installed in parallel with the normal exhaust system and the flow through it regulated by a valve, probably of the butterfly type.

The reason for using refrigerant gas instead of air for this thermally-regulated system is that it reduces the energy input required to adjust the height of the spring. That the reduction is considerable can be seen from the following example. In a spring about 20 cm diameter × 25 cm long, and with a rate of 100 kg/cm, the normal working temperature is about 30 deg C. If its static load is increased from 3 tons to 6 tons, and the deflection is to remain unchanged, an electrical energy of about 108,000 W-min must be supplied to raise the temperature of the air in the spring, and thus increase its volume the appropriate amount. On the other hand, if ammonia gas is substituted for the air, only about 8,800 W-min is required.

As is well known, ammonia gas has a very strong tendency to dissolve in moisture. Moreover, its properties vary considerably with the moisture content. From the accompanying curves, it can be seen that a rise in temperature of only about 20 deg C is necessary to double the pressure and therefore the load-carrying capacity. There are, of course, other refrigerant gases, the properties of which might be investigated.

Development work has been carried out by Waggonfabrik Uerdingen, in Germany, on a railcar equipped with gas-filled springs. Because of the relatively high temperatures that are liable to be attained in this kind of spring, metal bellows



Variation of pressure with temperature of various percentages of ammonia gas in water

were employed and they were covered with jackets of thermal insulation material. A disadvantage of this arrangement is, of course, the danger of fatigue failures of the bellows and possibily failures due to the corrosive effects of the gas employed. For road vehicle operation, there would also be some problems to be solved with regard to initial warming up immediately after starting from cold. Perhaps the corrosion and fatigue problems could be solved by the use of heat-resistant materials, such as silicone rubbers.

The development work confirmed that the spring rate obtained with this gas is lower than that which would be obtained with air. This is because, when the bellows extend, and the temperature of the gas is therefore lowered, some of the gas tends to condense into fluid droplets, thus releasing latent heat of vaporization; then, when the spring is compressed again, these droplets evaporate and absorb latent heat of vaporization. As a result of these alternate changes in state, the range of temperature variation between bump and rebound conditions is not so great as it otherwise would be.

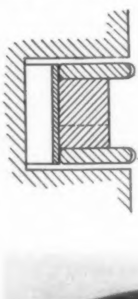
When the subject of air suspension is raised, the question invariably arises as to whether a hydro-pneumatic system, such as that employed on some of the Citroën cars, should be used. In this system, a jack containing an inert gas, separated from the hydraulic fluid by a rubber diaphragm, is employed as the springing medium. This gas is sealed in its container at the closed upper end of the jack, and adjustment for height regulation is effected simply by varying the volume of hydraulic fluid between the rubber diaphragm and the piston at the lower end of the jack. Thus, the column of hydraulic fluid acts simply as a strut, the length of which can be adjusted to regulate the static deflection.

It is, of course, generally undesirable to install both a hydraulic and a pneumatic pump; therefore, for commercial vehicles that are already equipped with an air compressor, the pneumatic spring is the obvious choice. However, where space is restricted, hydraulic actuation of the various components is in many instances preferred. Nevertheless, the pneumatic system has the advantage that its natural frequency does not vary so greatly with load as does that of the hydro-pneumatic system just described. Also, hydraulics generally involve sliding valves, and the system therefore has to be kept scrupulously clean.

Pneumatic systems, on the other hand, can be arranged in such a way that poppet valves with flexible seats can be used, and therefore extreme cleanliness is not of such vital importance. Sealing does not present so many problems in pneumatic systems and there need be no sliding pistons or other components liable to wear by abrasion. On the other hand, difficulties are liable to be experienced if precautions are not taken to prevent temporary failures due to the freezing of condensation round the valves and in the pipelines. To avoid these difficulties, some manufacturers simply incorporate a water and oil trap, while others introduce methyl-alcohol vapour into the system, to lower the freezing point of the condensed water vapour that inevitably is there.

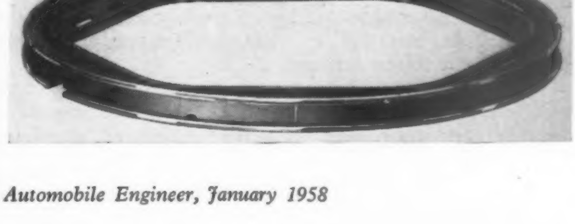
HEPOCROM OIL CONTROL RING

IN view of the success of chromium plating in reducing compression ring and cylinder bore wear, Hepworth and Grandage Ltd., of St. John's Works, Bradford 4, have developed the Hepocrom chromium plated oil control ring. This ring comprises four components, namely a spring expander ring, and an upper and a lower rail separated by a cast iron spacer. The outer faces of the two rails, where they bear on the cylinder bore, are heavily plated with high quality chromium. These rails are supported over their full radial depth by the cast iron spacer, which is accurately ground to control the working clearance of the ring in its groove. This prevents the rails from rocking, which otherwise might lead to oil pumping.



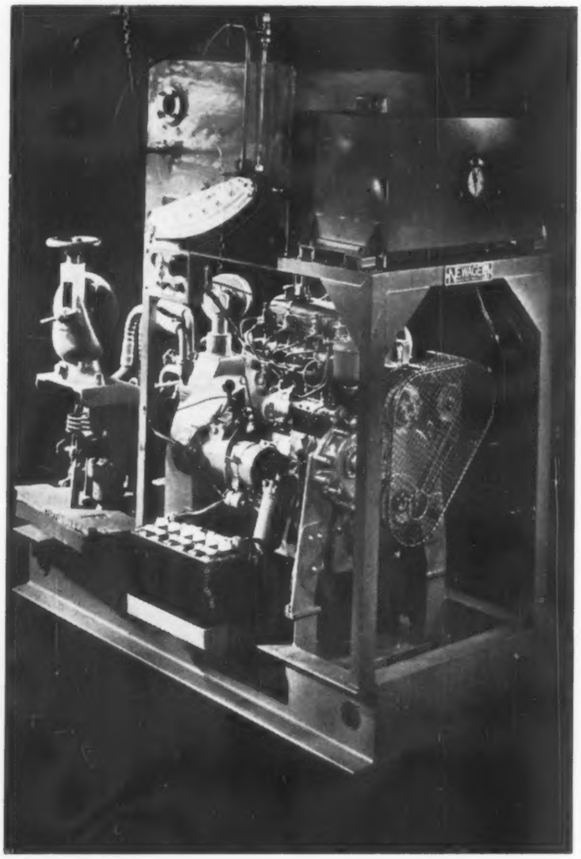
Cross section through a Hepocrom oil control ring, which comprises a spring expander ring and two rails separated by a cast iron spacer

The oil drainage groove in the end faces of the cast iron spacer ring can be seen in this view



Oil scraped off the bore into the space between the rails drains through the large opening at the gap, and also through grooves machined radially in the end faces of the cast iron spacer and passes away through holes drilled in the base of the ring groove to the sump. Since there are no spring rail-spacers, side drag in the groove is avoided and radial pressure between the rails and the bore is accurately controlled. Moreover, free movement of the ring in its groove enables it to follow readily the contours of a worn or oval bore, despite piston rock. Because of the moderate pressure of the ring on the cylinder wall and on the sides of its grooves, as well as the inherently low friction of the chromium plated bearing surfaces, reduction of mechanical efficiency owing to ring drag is minimized. These characteristics also lead to a low rate of bore wear.

The ring is designed in such a way that it cannot be fitted wrongly, provided the piston groove is the correct width and radial depth. There are no tapers, angles or special joints requiring attention, and no adjustments to make.



Engine Testing

Specially Designed Equipment for Technical Colleges and Training Establishments

Newage test equipment with 2·2 litre, high speed diesel engine. This equipment is designed as a unit, and incorporates all the necessary instrumentation for a wide range of tests

IN any heat engines laboratory, an essential item of equipment is an internal combustion engine incorporating a dynamometer and other testing and measuring apparata. The importance of such a unit has, in recent years, been greatly increased by the development of high speed, high efficiency engines. Only when students are instructed in the design features and characteristics of such engines, is it possible to ensure that their laboratory experience can later be usefully related to practice.

In the past it has been fairly common practice, particularly in the case of smaller engines, for college and other authorities to acquire the necessary equipment over a period of time, beginning probably with only an engine and dynamometer, then adding ancillary items when they proved indispensable. This procedure has obvious disadvantages. A test unit should be specifically designed for its purpose, thus ensuring that all components are correctly correlated and integrated to give maximum accessibility and adequate facilities for the performance of the various tests.

To help meet the need for thoroughly trained automobile engineers, Newage (Manchester) Ltd., 6 Carlos Place, Grosvenor Square, London, W.1 have developed a range of engine testing equipment. A typical installation is shown in the accompanying illustration. The standard equipment comprises an internal combustion engine, a dynamometer, indicator, air tank and fuel consumption meter, all combined into a single unit and conveniently mounted on a common baseplate. From the wide range of tests that can be carried out with this equipment it is possible to calculate efficiency at variable load, heat balance, torque, fuel consumption, etc.

Naturally, the most important item in the unit is the engine. It can be selected from the wide range of engines produced by the British Motor Corporation, and can be either a petrol or a diesel unit. For convenience and economy, the petrol engine usually selected is the 1,500 cm^3 model. Diesel units can be based on any one of the three Newage/B.M.C. diesel engines. The illustration shows a unit with a 2·2 litre diesel engine. The equipment associated with the petrol engine can be varied to suit individual needs.

Ignition. This can be either magneto or distributor. The latter is available with manually-operated advance and retard in addition to normal automatic mechanism. A scale and pointer are provided to facilitate readings of ignition settings relative to optimum position. The ignition system is suppressed to comply with statutory regulations.

Carburettor. Usually an updraught carburettor, with manually-operated continuously variable main jet, is supplied. Alternatively, a downdraught pattern can be fitted.

Cooling. This can be effected by header tank fed from the water main, or alternatively, and generally the more satisfactory method, by means of a separate water cooler and water circulating pump.

Starting. Electric starting is employed, with a 12 volt battery. For this type of set, the standard charging dynamo is usually omitted and a wall-mounted trickle charger introduced.

Silencing. Normally the system comprises an exhaust pipe and cylindrical silencer, but for applications where noise cannot be tolerated twin acoustic silencers can be fitted. They are usually located in a duct in the laboratory floor.

Coupling. All engines can have a small shaft extension on the end of the crankshaft to facilitate adaptation of a direct coupling drive to the indicator.

The dynamometer shown in the illustration is a Heenan and Froude hydraulic type, model DPX.2. It is mounted, together with the engine, on a rigid fabricated baseplate.

Normally, the drive between the engine and the dynamometer is by means of a cardan shaft, but where space is limited, a flexible coupling replaces the cardan shaft. Careful alignment and the provision of machined support pads on the baseplate ensure that no external loads are imposed on the dynamometer shaft and bearings. As in the case of the engine, the dynamometer may be cooled by mains water or, preferably, by means of water circulated from a separate cooler.

The indicator is bolted and dowelled in position on the baseplate. It is driven directly from the engine through a vernier flexible coupling. This permits easy removal of the indicator for use elsewhere, and also provides maximum accessibility during normal operation with the engine. Dobbie and McInnes "Farnboro" type indicators are normally used, but alternative types can be fitted when required.

Provision is made to ensure that the engine T.D.C. position can easily be found and marked on the diagram sheet, together with calibration lines at any desired pressure reading. Light pressure work is facilitated by a low pressure unit which, when required, is attached to the standard indicator on the side opposite to that occupied by the normal pressure equipment. The electronic relay unit is mounted below the fuel tank in a position conveniently adjacent to the indicator. It can easily be removed for routine inspection.

Compressed air for use with the indicator is stored in a compact pressure cylinder mounted alongside the base. Under normal conditions the cylinder has sufficient capacity to produce 50 diagrams. For light pressure work, an electric motor driven pump is available. This provides the necessary low pressure air and, when required, also operates a vacuum pump.

Generally the indicator is arranged for use with only one cylinder of the engine. However, when required it can be connected to all cylinders, the pressure pipes being routed through a distribution block, an arrangement that facilitates the change-over from high pressure to low pressure work. A multi-way switch enables the impulse from any selected cylinder to be passed to the relay unit. The pick-up units are usually mounted to the cylinder head by means of a water-cooled adaptor that screws into the cylinder head in the position normally occupied by the spark plug, which is re-positioned in the side of the adaptor. Although this method of adapting the pick-up unit necessitates re-positioning the spark plug, it does not, in practice, introduce any undesirable effects.

For air consumption measurement the inlet air is taken through a calibrated orifice plate into an air box before passing to the engine. This air box is of sufficient size adequately to damp undesirable pulsations. It is covered with a coating of flexible material to reduce noise to an acceptably low level. Air flow measurements are taken direct from a Kent curved tube manometer, which is usually mounted on a wall or placed on a stand near the engine unit. Alternatively, where it is desired that experiments be carried out on the actual measurement of air flow, a U-tube can be fitted to the side of the box. As the U-tube involves considerable calculation to obtain the air flow figures, it is only supplied when specifically ordered.

An Amal flow meter is used for fuel consumption on a petrol-engined unit. It provides instantaneous and accurate figures for fuel consumption at any speed or load. A diesel unit is supplied with a glass cylinder with markings to indicate 0·5 and 0·25 pints. This, in conjunction with a stop watch, provides a simple means of computing fuel flow. Fuel cocks are provided to allow a continuous series of fuel tests to be carried out without stopping the engine.

All normal engine instruments and controls are provided. These, including a variable speed cable, are conveniently grouped on an instrument panel. Special care has been taken to reduce vibration and noise to a low level. All fuel lines incorporate flexible inserts, and the fuel tank is flexibly mounted on the supporting framework. To assist students in the operation of the engine and ancillary equipment, labels giving the basic operating instructions are permanently attached to the unit at the appropriate points.

Star Tolerance Rings

As can be seen from the accompanying illustration, a Star tolerance ring consists of a spring steel strip containing a number of corrugations. Each corrugation acts as a spring, and when compressed between two machine parts exerts a constant load dependent upon the degree of compression. The straight ribbon on either side of the corrugations prevents the pitch of the corrugations from changing on assembly; the shape of each corrugation merely alters.

The radial space between the mating metal components into which the tolerance ring is to be fitted is so determined as to compress the corrugations, with the result that a predetermined degree of torque can be transmitted. To transmit greater or lesser torque is simply a matter of adjusting the width of the tolerance ring.

There is a wide range of applications for which Star tolerance rings provide many technical and economic advantages:

(a) They provide an interference fit with wider limits on both machine parts.
(b) They transmit a given amount of torque.
(c) They compensate for heat or work expansion.
(d) They compensate for variations in machining tolerances and automatically adjust for a degree of misalignment.

In addition to allowing wider machining tolerances on mating parts that must be an interference fit, Star tolerance rings can also be used to advantage in some simpler applications, such as the fitment of hand wheels and lever knobs which are normally fitted by screw threads. The cutting of screw threads can be completely eliminated, and the components can be securely held by the press fit afforded by the tolerance ring.

A further advantage is the ability to take up differential linear co-efficients of expansion of different metals, such as in the case of a ball bearing fitted in a light alloy housing. Star tolerance rings have been used on the Continent for years in many branches of engineering. They are now available in Great Britain from George Angus and Company Limited, Oil Seal Division, Coast Road, Wallsend-on-Tyne.

Typical Star tolerance rings

New Plant and Tools

Recent Developments in Production Equipment

THREE standardized, interchangeable head units, for boring, grinding and measuring respectively, are available for the optical co-ordinate machine recently introduced by the German firm of Friedrich Deckel. Thus, where full utilization is assured, the use of a single attachment provides an efficient single-purpose machine. Conversely, the range of attachments converts it into a general-purpose machine adaptable to a wide variety of work.

In general construction the machine has a rigid box-type base and the movable units, the co-ordinate worktable and gearbox, are adjustable in guide ways by means of push-button controlled, rapid traverse motors and manually operated fine setting controls. Settings for the worktable over an area of 15·7 in × 10 in, are easily and directly read from screens with co-ordinate measuring recticule, providing an accuracy of adjustment to 0·00012 in. The gearbox is arranged for vertical movement and carries a bracket to receive the head attachments. This bracket is designed to compensate automatically for the effects of thermal expansion. All controls are within easy reach of the operator and the electrical equipment is housed in a separate cabinet.

Boring machine LKB. The boring head is a standardized unit that is already proved as an attachment for the tool milling machine manufactured by the Company. It can be tilted through 90 deg and has a range of twenty spindle speeds from 40 to 3,150 rev/min. The maximum height of work that, as standard, can be accommodated is 19·9 in, but this figure can be increased to 24 in by means of a raising unit. Spindle travel, with both manual and power operation, is 3·94 in.

There are six different feed rates from 0·0004 to 0·005 in/rev, and all can be used with any of the available spindle speeds. Capacity is drilling up to 0·59 in diameter and boring up to 6·9 in diameter. The area of the worktable is 25·6 in × 11·8 in. A full range of accessory equipment, including a universal facing and boring chuck in two sizes, is available.

Grinding machine LKS. A high-precision grinding head, which can be tilted 45 deg to either side of the vertical, enables grinding operations to be performed to extremely close limits. The range of applications includes cylindrical and tapered, internal and external surfaces; straight, angular, and segmentally shaped grooves; and the form grinding of profiles composed of radii and tangents. Stepless variation of spindle speeds—15,000 to 30,000 rev/min for wheels up to 22 mm diameter and 30,000 to 60,000 rev/min for wheels up to 15 mm diameter—is by a motor-driven frequency changer housed in the separate switch cabinet.

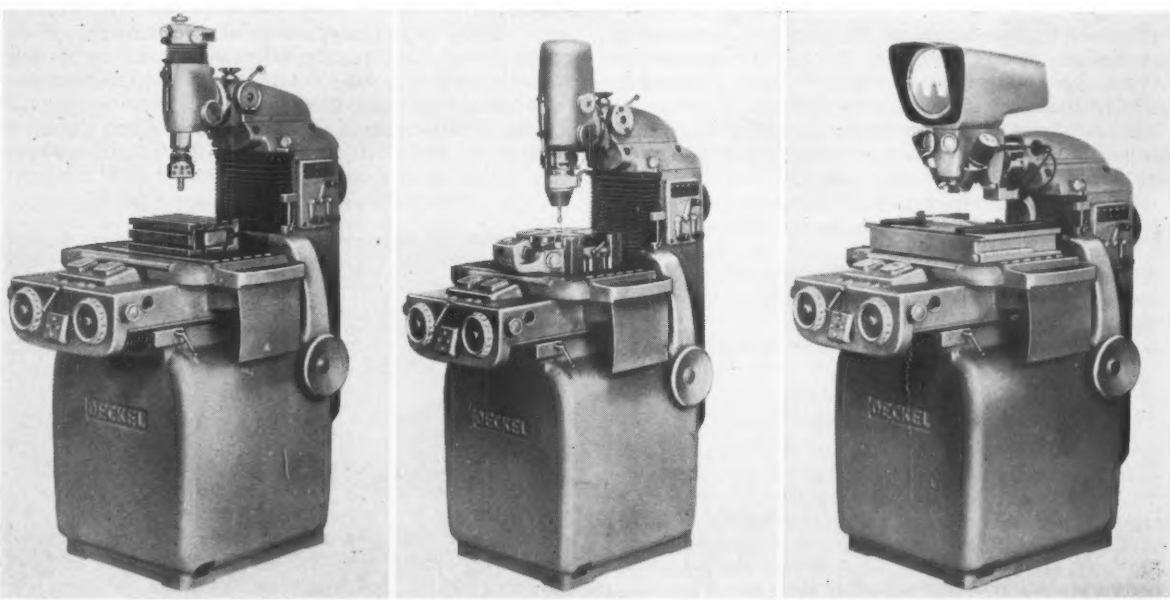
Spindle travel is from 0·156 in to 3·15 in and the oscillating vertical movement has three feed rates, 0·12, 0·24, and 0·48 in/rev. The number and progression of the spindle planetary speeds are 12 and 1·25, giving speeds from 40 to 500 rev/min. Minimum and maximum diameters of bores that can be ground are 0·031 in and 4·9 in.

A 12 in diameter circular table with optical adjustment of polar co-ordinates is shown on the grinding machine illustrated. It can be used either horizontally or vertically and readings down to 1 sec of arc are taken on a square screen. Indexing accuracy is 5 sec of arc.

Measuring machine LKM. Fitted with the measuring projector. and used in combination with a variety of attachments and accessories, the machine can measure and check contours, the shape and pitch of screw threads, the accuracy of form tools, templates and the like. The magnification of

Three versions of the Deckel optical co-ordinate machine. Interchangeable heads can individually provide single-purpose boring, grinding, or measuring machines or, in combination, a general-purpose machine

Burton, Griffiths and Co. Ltd.



×10 or of ×20 diameters is fixed and remains constant. A vertical travel of 3·15 in is provided for the measuring head, which can be tilted 15 deg on each direction about the vertical optical axis. The diameter of the projection screen is 7·9 in.

Deckel machines are factored by Burton, Griffiths and Co. Ltd., Mackadown Lane, Kitts Green, Birmingham, 33.

"Multisize" collets

Substantial economies in production equipment can be achieved by the use of these new precision collets, it is

Each "Multisize" precision collet has an operational range over 0·125 in on diameter

F. Burnerd and Co. Ltd.

claimed. Each collet has dimensional flexibility to the extent of 0·125 in on diameter and a range of eleven units, already available, can handle work diameters from 0·125 in to 1·5 in. Other ranges will be introduced in the near future by the manufacturers, F. Burnerd and Co. Ltd., 5 Balfour Place, Park Lane, London, W.1.

"Multisize" collets ensure exceptional accuracy in centring and a grip many times greater than the conventional spring collet. The grip is provided by the action of accurately ground, wedge-shaped blades radially spaced in the tapered body of the collet. Depending upon the size of the collet, six or eight blades are fitted. The unit is not subject to deformation since the blades are free to slide within the slots in the body and thus to accommodate the diameter of the bar stock. When closed, the gripping action of the blades is exerted parallel to the bar over their entire length. It is this movement that gives the collet its exceptional accuracy. Deep cuts can be taken on work without risk of slip or chatter. Accuracy in machining is claimed to be within 0·0005 in at a distance of four inches from the nose of the collet.

Using conventional equipment, from 16 to 20 collets may be required to handle all diameters, including metric sizes, from 0·125 in to 0·25 in since deviation in excess of, say, 0·004 in leads to uneven grip. A single "Multisize" collet can thus replace a considerable number of standard collets.

A collet chuck, illustrated on the Contents page, has been designed specifically for use with the new collets. It can be supplied with a flange for attachment to an adaptor plate or for direct mounting on various standard spindle noses. A feature of the chuck is a ball thrust race which virtually eliminates operational friction. Time to change from one collet to another is no longer than with ordinary equipment.

Accurate length positioning of short-length components, as for second operation work, is arranged by means of an adjustable end-stop. It is readily fitted or removed as it is a push fit in the chuck body. Ball detents engaging in a recess machined in the body locate the stop in position and a central screw provides adjustment.

Erodosharp Mark II

The first Erodosharp machines for re-servicing carbide tools were introduced by Wickman Ltd., Banner Lane, Tile Hill, Coventry, about six years ago. Since then many of these machines have been in continual use for single-point tool dressing with eminently satisfactory results. The Erodosharp Mark II, a development of the earlier machine, embodies two major refinements in design, based upon the experience gained. Two revolving electrodes and adjustable tool rests are now provided, and an extractor fan is built into the machine base to enable fumes produced during processing to be ducted to atmosphere independently of other extractor systems.

In appearance it resembles a conventional grinding machine, except that two revolving iron electrodes replace the usual abrasive wheels. Metal particles are removed from the tool by the effects of a controlled electrical discharge between electrode and tool across a dielectric gap, which is a thin film of oil on the revolving electrode. Since there is no physical contact between electrode and tool, there is no load on the wheel, no wear due to abrasion, and no heat generated in the tool. This means that the tip and shank of carbide tools can be dressed simultaneously without any risk of the tip cracking from thermal shock. The conventional method of using different grinding wheels for roughing and finishing is eliminated. Finish is selected simply by operation of a three-position switch, which gives different cutting rates for roughing, normal stock removal, and finishing.

The actual tool dressing process is performed in the same way as on grinding machines, except that no pressure is required, and re-servicing times are closely comparable with those of conventional methods. Tools dressed by the Erodosharp process have been found to have a serviceable life

The Wickman Erodosharp Mark II machine for re-servicing carbide-tipped tools

Wickman Ltd.

as long as tools ground by conventional methods, and in some cases much longer. This is because the finish on Erodo-sharpened tools consists of minute saucer-shaped depressions which resist abrasion better than the minute ragged indentations left by ordinary grinding. Wheel life is almost indefinite, as the slight pitting resulting from prolonged use can easily be smoothed off with an abrasive stick.

The largest single economy offered by this method is the direct and immediate saving in diamond wheel costs. Except for the lapping of superfine cutting edges, the Erodosharp will produce all the results achieved with diamond wheels.

Precision micro-drilling machine

Accurate chucking and the maintenance of drill concentricity are prime considerations in the successful drilling of holes in the micro-drill range. The Dumore Co. of Racine, Wisconsin, U.S.A. has recently developed a versatile, sensitive-feed, micro-drilling machine with an accurately centring magnetic chuck. A drill is mounted in an appropriately sized collet and can then be inserted in the magnetic chuck and centred without the need to stop the spindle rotation, thus facilitating a rapid changeover when drilling successive holes of different sizes.

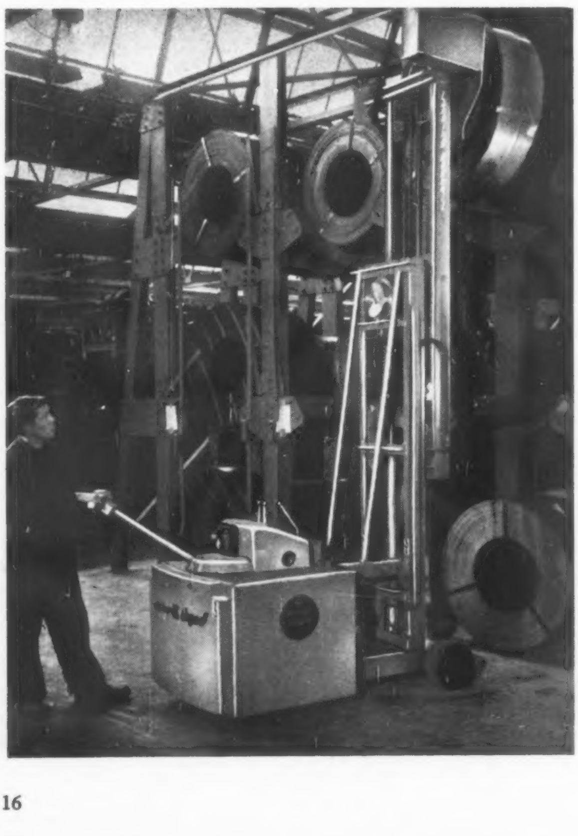
The worktable, drilled and tapped for the attachment of fixtures, .s provided with longitudinal and transverse movements up to 1·125 in. Control is by conveniently positioned knobs and the extent of movement is indicated on large-diameter Ames dial gauges, permitting rapid location and precise spacing of holes. A weight, adjustable on a lever arm, counterbalances the weight of the table and fixture to permit sensitive feed of the table by means of a large control knob. Table feed is limited by a calibrated micrometer positive stop and it is possible to maintain a tolerance of 0·0005 in on the depth of holes drilled.

Voltage control by means of an adjustable transformer

Sensitive feed micro-drilling machine
Dumore Co.

is used to vary the motor speed from 2,000 to 17,000 rev/min. While operating, the progress of the drill can be watched by the aid of a mounted $\times$ 10 power magnifier focused on the hole.

Lift-truck adapted for coil handling
Lansing Bagnall Ltd.



Handling and storing coiled steel strip

Even where relatively small quantities only are involved, the storage and handling of strip steel in coils is likely to present problems. The stock should be readily accessible, it should not be liable to sustain mechanical damage, and there should be no "dead ends" in the storage area where coils may remain for long periods and possibly suffer deterioration.

Lansing Bagnall Ltd., of Basingstoke, made an extensive study of the problems for a large production plant and designed and developed a racking system and also a special coil-handling head attachment for a lift truck of their manufacture. This attachment, replacing the usual fork members, consists of a large vertical face plate from which projects at right angles a hollow spigot or core. The outer diameter of the core is slightly less than the inner diameter of the smallest coil of steel to be handled. On the underside of this core is a segment, subtending an angle of about 60 deg, which is displaceable vertically downwards on guide rails mounted on the face plate. When the core has been entered into a coil of steel, the segment is hydraulically operated and brought into engagement with the lower run of the inner diameter of the coil. Thereafter, the coil is securely held by hydraulic pressure until the operator has transported the load and positioned it, when he releases the segment and disengages the core.

The three-tier racking structure comprises eleven rows of columns defining ten passages or channels. On each side of a channel the columns are bracketed to carry, at two levels, inclined racks to receive the coils. At floor level, guide rails serve to align and locate a third row of coils. The truck can traverse the entire length of each channel and deposit or withdraw coils from any of the three storage levels.

Although the racking structure occupies a stores area of only 1,600 ft², it has sufficient capacity to hold more than 400 tons of steel strip in stock and readily available.

PLASTIC-COATED STEEL

New Material by John Summers & Sons Ltd. can be Bent, Formed, Drawn, Seamed and Welded

A REMARKABLE new coated sheet steel, termed "Stelvetite", has recently been introduced. Developed by John Summers and Sons Ltd., of Shotton, Chester, in collaboration with B.X. Plastics Ltd., it is a strip-mill, cold-reduced steel with a specially formulated Velbex P.V.C. coating. The reverse side of the sheet has either a Bonderized surface or an electro-zinc coating.

Stelvetite is produced on a continuous strip mill in a new plant and is at present available in all gauges from 26 B.G. to 16 B.G. (0·0196 in to 0·0625 in). Sheet width is in various dimensions up to 48 in and sheets are supplied in lengths up to 144 in. The thickness of the P.V.C. coating is 0·014 in. This figure was adopted as the optimum dimension after an exhaustive investigation of all foreseeable requirements and of the practical economic aspects. The standard coating thickness gives adequate protection from atmospheric corrosion, electrical insulation up to 4,000 volts, within wide limits is resistant to acids and alkalis, and is resistant to oils and greases.

Other features are good resistance to abrasion, stability at temperatures higher than those normally associated with P.V.C. alone, and it will not support combustion. It is warm and pleasant to the touch and can be washed and cleaned with soap or most detergents. Small quantities are available in one of four standard colours—white, green-grey, primrose or scarlet—but for larger quantities from minima of 10 tons in 26 B.G. to 30 tons in 16 B.G. sheet, any of the colours available for P.V.C. material can be supplied. The standard surface finish is a neat embossing, termed "Seal" as it simulates the surface of pin-seal leather. At present only a limited range of other embossings is available but either a smooth finish or a hide finish can be supplied if ordered in substantial quantities.

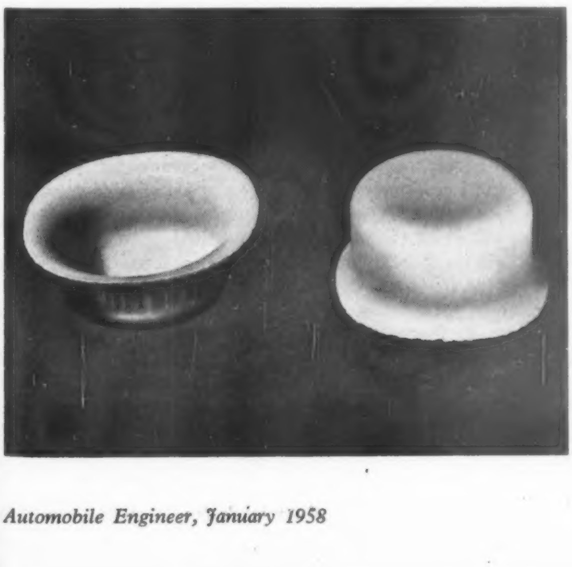
The outstanding property of Stelvetite is that it can be worked and treated in production as ordinary sheet steel without damaging the coating. Using ordinary tools it can be formed or bent with the coating either internal or external. It can be drawn in the usual dies and a test cup 5½ in diameter has been drawn to 3½ in deep with a completely unimpaired external coating. In this connection it is reported that pressing or drawing operations are actually facilitated by the P.V.C. coating, which possesses a certain resilience and also exercises what may be termed a dry lubricating function. As will be seen in the illustration of the test cups, both produced with the standard press tools, the one with the coating inside exhibits draw markings on the external steel surface while on the other the external coating, which is "seal" embossed, is unblemished.

As an example of the tenacity of the bond between the P.V.C. coating and the sheet steel, the illustration of the "expanded" panel is included. In this the material is pierced at a close pitching, pressed out, and then bent by stretching, but in no instance is there any evidence of lifting or detachment of the coating.

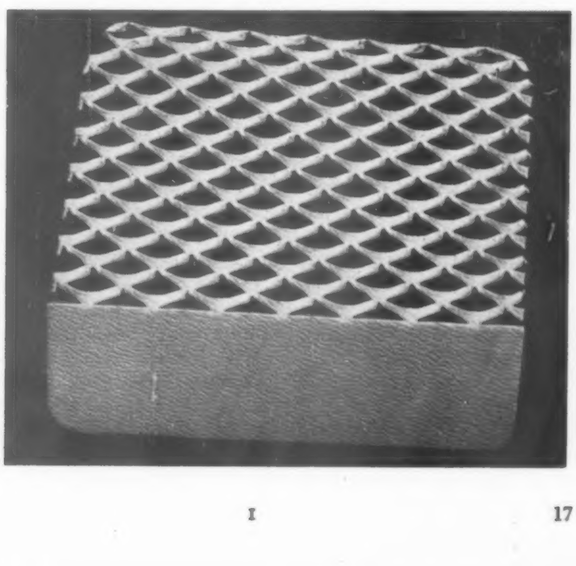
Panels can be secured together by spot-welding at the marginal flanges provided the P.V.C. coating is first removed. This is effected simply by running over the flange with a heated iron furnished with a shouldered bit by which it is guided from the flange edge. Alternatively, the appropriate width of the flange could be cleaned by "wheeling" the panel between heated rollers. For lightly stressed structures, panels can be joined if the P.V.C. coatings of abutting flanges are welded together by normal plastics high-frequency welding equipment. Any attachment or fitting of plastics, a decorative motif, badge, name-plate or the like, can be permanently affixed to the coated surface of a panel by the same method. Similarly, for a detachable panel the cut edge may be closed by a channelled strip of P.V.C. or a decorative bead or moulding. An application of particular interest for vehicle interior trim, an inner door panel for instance, is the method of attaching an upholstered or padded sheet of P.V.C. directly to the coated panel by H.F. welding around the margins. Decorative material of this type is available in a wide range of colours, embossings, and patterns.

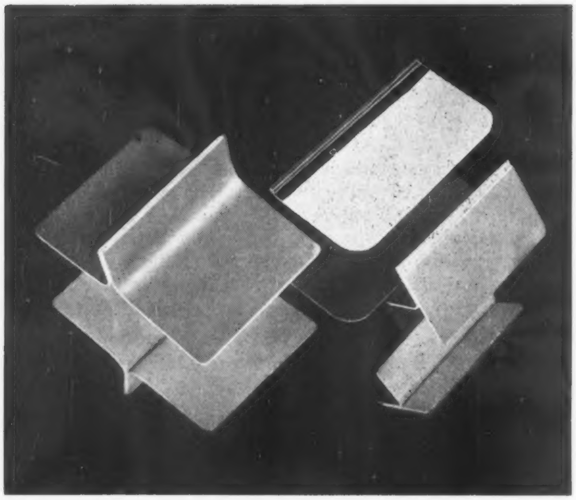
The securing of metal fittings or means of attachment to the metal surface of Stelvetite can be readily effected by

Test cups illustrating the "drawability" of Stelvetite sheet. No draw marks appear on the externally coated example

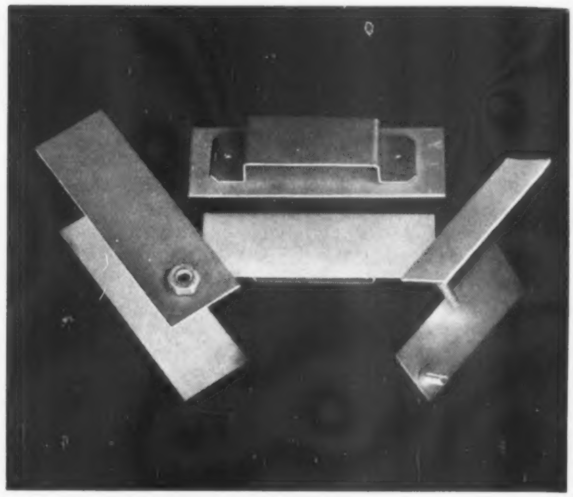


A specimen showing the adhesion of the P.V.C. coating under exacting production conditions





Assembly of fabricated parts by H.F. welding of the P.V.C. coatings. Seams can be filled with P.V.C.



Projection-welded nuts and studs. Mirror view to show unaffected P.V.C. coating

the method of projection welding without in any way damaging the coated external surface. An illustration shows examples of this technique; the articles are placed on a mirror so that both sides can be observed simultaneously.

Since the coating is resistant to acid it is possible to electroplate either the metal surface or cleared areas or margins of the coated surface, should this be desired for either constructional or decorative purposes. The only limitations in the process are that the aqueous plating solution should be of not more than 30 per cent acidity and that the operating temperature of the solution should not exceed 150 deg F. Providing these limits are observed the P.V.C. coating will be unaffected.

One aspect of production, of some importance, is that press trimmings, cut-offs, and waste Stelvetite relegated to scrap do not need to be segregated. Since P.V.C. is a carbonaceous material it has no deleterious effect and can be sent to the furnace with the normal steel scrap. Thus existing methods of scrap collection and disposal require no modification or reorganization.

Although it is more costly than ordinary sheet steel, Stelvetite eliminates the need for surface finishing treatments and thus shows an overall economy. Production plant, material, time, and labour can be saved. It is cheaper, easier to work and does not require the finishing of stainless steel.

Although a large scale plant has been brought into operation and production is in quantity, supplies of Stelvetite are not unlimited. Already they are being drawn upon by an extensive range of industries, all potentially large consumers. For the motor industry the immediate applications are for interior furnishing and trim; door panels, seat backs and facia boards are obvious examples. The potentialities of the material for the main body shell, however, should not be overlooked. Necessarily, a complete shell would be a later development, since a body would need to be designed specifically for fabrication in the new material to secure the optimum advantage.

Somewhat more care would be necessary in handling in the shops, but the colourful, finished appearance of the sheet stock—it is supplied clean and interleaved with paper—creates a good psychological impression. It has already been noted that operatives refrain from walking on Stelvetite sheets; an all too common practice that cannot be completely eradicated when the usual oiled sheet steel stock is used. Certainly cleaner working conditions will be possible in the press shop.

Constructionally, no major problems are envisaged. It is a definite advantage that conventional tools and equipment can be used, as with ordinary sheet steel, and familiar production methods followed. The problem of protection against corrosion, atmospheric or electrolytic, is no worse than with ordinary steel and, particularly in the case of the electro-zinc coated sheet, may be considerably easier of solution. Cut edges may be folded or rolled to enclose the edge or a P.V.C. moulding or extrusion can be used to seal it. Seams between panels can be filled with P.V.C. and smoothed or radiused to present an unbroken surface by a method closely resembling the solder filling of seams in ordinary steel bodies. Hot-gas torches, either acetylene gas-heated or electrically heated types, are used with a

Upholstered P.V.C. sheet attached to P.V.C. coating of Stelvetite panel by H.F. Welding

filler rod of a composition identical with that of the coating. The heated gas may be compressed air or nitrogen. Suitable torches and equipment are manufactured by Rediweld Ltd, 15-17 Crompton Way, Crawley, Sussex.

Since Stelvetite is supplied as a finished material it should be treated as such in the workshop. Research, development and testing of the new material have been conducted over a period of years and extensive experience has been accumulated on methods of handling, working, and fabrication. Manufacturers are invited to consult with the producers to ensure that the best possible results are achieved. In any circumstance it is advisable to state the application for which Stelvetite is intended so that the correct grade of steel and the most suitable formulation of plastic can be supplied.

Engine Lubrication

Tests on a Compound Based on Molybdenum Disulphide

IN the past few years the lubricating properties of molybdenum disulphide (MoS_2) have been the subject of many investigations and tests which have proved that this solid lubricant has very valuable properties, particularly in applications where pressures and temperatures are high. Some recent tests, which are described in these notes, are of special interest to automobile engineers. They were concerned with "Auto-Moly," an MoS_2 engine compound developed by Charham Products Ltd., 5a Market Place, Acton, London, W.3, for use in internal combustion engines.

Auto-Moly engine compound is made from the purest refined molybdenum disulphide. Its average particle size of under one micron is suitable for passing through any normal filter. It is an almost inert substance and is completely non-toxic. The purpose of the tests was to determine whether the surfaces formed after Auto-Moly had been added to the oil were strong enough to permit the engine to run (a) in conditions of oil starvation, and (b) without any oil at all.

The test vehicle was an Austin A50 Countryman van with a factory-reconditioned engine. It was run in on Shell oil to which Auto-Moly had been added. The big-end bearings were of lead-indium, and the main bearings of babbit white metal. After 872 miles, the engine was stripped and inspected by Milne and Russell Ltd., of Croydon.

Condition of engine after running-in

Inspection of the cylinder walls and pistons showed that a very useful bonded coating of MoS_2 had already been applied to metal surfaces. The usual bright, highly polished surface finish was observed to be complete throughout all units. It was noted that the small amount of carbon deposit present on the top of each piston was adhering very loosely and flaked off almost at a touch, leaving a clean MoS_2-coated surface below. This, it is felt, is evidence of the cleansing action of Auto-Moly in limiting the formation of carbon deposits.

Valves, valve guides, stems, big-end side faces and the faces of the crankshaft webs were all well coated with MoS_2 and showed polished surfaces. There was also some evidence of MoS_2 adhering to the bright surfaces of the white metal bearings. For later comparison, all wearing parts of cylinders, piston rings, crankshaft, bearings, etc., were accurately measured.

In order to provide accurate data on engine performance, the following additional instruments were fitted.

(1) A Redex vacuum gauge to indicate engine performance through negative boost readings (fitted on top of manifold).
(2) Oil temperature gauge with connection to the bottom of the oil sump to give an indication of operating temperatures below the crankshaft. Readings in air instead of liquid to be computed above indicated temperatures.
(3) Oil pressure gauge.
(4) Radiator temperature gauge.

Approximately 5 oz of Auto-Moly were added to the gearbox and back axle. It was seen that the gearbox, though recently topped up, was practically empty. This was due to a faulty seal, which was not discovered until considerably later.

To complete the running-in over a sustained period at high speed, and to test the performance of the vehicle, a run was made to Exeter on the A30 road. During a prolonged run at speeds from 60 to 80 m.p.h., the temperature gauges were both allowed to reach maximum, 212 deg F, and stops were made to check oil and water levels, and exhaust and idling performance.

Boost level readings were noted for different throttle settings, and, as far as possible, fuel consumptions were determined for various speeds. It was found that for a reading within the recommended gauge limit of 18-20 in, the economical speed/consumption level was approximately 42 m.p.h. The actual readings were:—

40/65 m.p.h.	28 m.p.g.
40/45 m.p.h.	32·5 m.p.g.
30/35 m.p.h.	30 m.p.g.

These figures were required for comparison with later readings obtained during the oil-free run.

Test 1—Oil starved conditions

For this test, the engine sump was first drained and then approximately $1\frac{1}{2}$ pints of Auto-Moly were put into the sump. For the next 300 miles—Torquay, Mevagissey, Crediton—the vehicle was generally driven at normal speed, and at times at high speed, with an oil pressure reading of approximately 10 lb/in², as against the average of 60 lb/in² when running on a full sump, to check temperatures and general engine behaviour.

Apart from oil pressure, no alterations were observed in instrument readings, performance, acceleration and fuel consumption. The readings at average cruising, 40-45 m.p.h. were:

	Sump full	Sump with $1\frac{1}{2}$ pints
Boost	18-20 in	18-20 in
Oil pressure	60 lb/in²	10 lb/in²
Oil temperature	160 deg F	160 deg F
Radiator temperature	165 deg F	165 deg F

Test 2—Oil-free run

On 17th August, the oil sump was completely drained, approximately 1 pint was removed, at Crediton, Devon. The vehicle was then driven to Exeter and thence by A30 to London. Short stops for refreshment were made at Shaftesbury and Salisbury. The log for this journey is shown at the top of the next page. Boost was 18-20 in and the oil pressure nil throughout. Readings were taken at 30 m.p.h.

In the circumstances, the vehicle performed very well. There was no observable difference in performance apart from suspected sluggishness on longer hills, which, however, may have been psychological. With increased throttle, the boost behaved normally in dropping back, so no engine labouring could be established. The valve or bearing noise at low throttle became fairly continuous but was not excessive, since every care was taken to drive outside this trouble zone. The boost pressures were unaffected at 30 m.p.h. average cruising. Knocking was not apparent at idling speeds nor was any excessive smoke apparent at stopping points either from the exhaust or the oil filler cap.

Speedo-meter reading	Oil temp deg F	Radiator temp deg F	Remarks
62,696	120	160	Valve or bearing knock at low throttle
62,712	128	165	Still knock on low setting; free above and below
62,716	130	168	Speed acceleration the same as before
62,720	135	166	No change
62,725	140	164	No change
62,730	140	166	Practically no knock
62,740	140	166	Knocking at low setting
62,750	142	166	Same as before
62,770	140	168	Same as before
62,800	140	166	No change
62,805	140	170	More pronounced big-end noise
62,820	140	165	Knocking only at low throttle. O.K. above and below
62,850	138	165	Still knocking at low setting
62,900	140	166	Same

Second oil-free run

At a speedometer reading of 62,906 miles, the petrol tank was filled with three gallons of premium grade petrol and the vehicle was driven to Moons Garage, Victoria, where the sump was drained. This test consisted in a run to Brighton without oil. The test started at a speedometer reading of 62,910 miles and finished at a reading of 63,010 miles. At this point, the petrol level was the same as at the starting point. The average consumption was, therefore, slightly over 30 m.p.g., which is similar to the original finding.

It is interesting to note that on the 18th of August, after the oil-free run from Exeter, the engine started more easily than was normally the case. During the oil-free run to Brighton, vehicle performance was in every way similar to that of the run from Exeter to London. Temperatures, pressures, acceleration and behaviour were normal. Excess knocking did not become apparent, although it must be noted that care was taken to drive outside the limits of the knocking zone on the throttle.

To conclude the test, the vehicle was driven locally in London for demonstration purposes, without oil in the sump. At a mileage of 63,191, the vehicle was returned to Milne and Russell Ltd., for stripping and examination.

Test Results

A glossier and more evident bonded coating of MoS_2 was apparent on the cylinder walls and there were no discernible marks of scoring. The pistons were unmarked, and the rings were highly polished and the ridging left after manufacture was still in its original state. No measureable wear had taken place.

Camshaft, valves, valve guides, stems. These showed no change and no measurable wear. All the components were in a clean state with no signs of sludging.

Connecting rods. The small-end bearings showed no observable signs of wear. Each lead-indium big-end bearing shell had, to a greater or lesser extent, lost part of its soft facing. In the case of Nos. 1 and 3 bearings, the facing had been removed somewhat unevenly, particularly about the oil hole. This wear can be assumed to be due to hammering without the normal cushioning effect of an oil film. It could no doubt be reduced or eliminated by the fitting of solid phosphor bronze bearings with a closer clearance. From a depth point of view, the overall wear was only slightly more than the usual limits expected for a normally lubricated bearing over a similar working period.

Crankshaft and main bearings. Some evidence of scoring was apparent on Nos. 1 and 3 bearings at points immediately opposite each oil hole in the bearings shells where the facing had been removed unevenly. The depth of wear was, however, well within normal limits. The main bearings and shells showed no observable signs of wear and were even throughout the area of each surface.

Observed measurements

Cylinder bore wear, in				
	No. 1	No. 2	No. 3	No. 4
Measurement before running	2·6085	2·6085	2·6085	2·6090
Measurement after running	2·6090	2·6090	2·6090	2·6100

Crankshaft wear, in		
	Before running	After running
Main journal measurement	1·981	1·979

Crank pin measurements before running		1·856
Measurements after running	No. 1	1·851
	No. 2	1·852
	No. 3	1·852
	No. 4	1·852

It is almost unnecessary to say that these tests were not run to show that lubricating oil can or should be dispensed with. The primary aim was to prove that Auto-Moly, when added to the lubricating oil of an internal combustion engine, is rapidly and evenly dispersed throughout the engine, and because of its natural attraction, bonds to the metal surfaces to form a layer of immense and lasting bearing capacity, which reduces friction and gives added lubricant protection.

Counterbalanced Push Travel Truck

ECONOMY of operation in materials handling for short distances is claimed for the TPS MachinLoder recently developed by Trucks and Pallets (Stackers) Ltd., Gas Street, Birmingham, 1. This truck has been developed to replace high-priced, power-driven equipment. It has extremely low traction resistance and is simple to move and manoeuvre by means of positive tiller steering. The truck is counterbalanced and of very small size so that it can be used in restricted spaces where larger equipment would be completely unsuitable.

The design is such that the truck can be placed flush with a machine without obstruction at the base. As a consequence the loading of lathes, presses, milling machines, and process plant is greatly simplified. A 4 ft 6 in lift from floor level is obtainable by manual or battery electric means. The standard rating of 500 lb at 12 in centres gives a safety factor of approximately 80 per cent. Alternative ratings of 750 lb and 1,000 lb are available.

The TPS MachinLoder is built from high quality materials and components with ball and roller bearings throughout. The hydraulic system is sturdy and thoroughly efficient. In addition to being used for loading, it can be employed equally effectively for pallet handling in confined spaces. With additional fittings it can be adapted for many other applications and operations. Removable 24 in forks are fitted as standard. The optional extras are:—

(1) 20 in jib
(2) Squeeze clamps
(3) Roller platform

Other versions of the TPS MachinLoder are available, with load capacities of 1,000 lb and 2,000 lb at 24 in centres with lifts up to 6 ft.

Safety Belts

Details of a Device to Minimize Injury

MODERN coachwork, particularly when composed entirely of steel and the unitary method of construction is used, is extremely strong. It offers a high degree of protection in the event of accident, but this in itself is not enough to ensure freedom from injury to the car occupants. It has been proved that a car may somersault several times without very serious damage, but obviously under these conditions those inside the car would undoubtedly suffer injury. In recent years a number of ideas have appeared to improve the safety of those seated in the car, and quite frequently the padded windscreen facia has been used, this giving a pleasant finish, quite apart from its efficiency. In some countries, particularly America and Sweden, interest has been shown in the safety belt, an idea obviously inspired by the similar device that is used in aircraft.

The provision of a safety belt is not perhaps quite so simple as it may sound, for little thought is needed to determine that however strong the belt in itself may be, its efficiency will be rendered neglible if the attachment to the car is weak. A very close study of this matter has been made in Sweden, where from accident statistics it was proved that the driver, of all the occupants in the car, had the greatest protection in the event of accident, only 6 per cent of drivers being injured. His fellow front-seat occupant ran the greatest risk with a figure of 70 per cent, while the two rear seat occupants each carried a figure of 12 per cent.

As a result of these studies and in conjunction with aircraft authorities, a Swedish safety belt was devised and is now being offered in this country by Delaney Gallay, Vulcan Works, Edgware Road, London, N.W.2. It is made in varying styles to suit the different types of seat now being used, but its salient feature is the provision of shoulder straps in addition to the waist belt. A waist belt would hold a passenger to the seat but in the event of a severe collision the shoulders would be thrown forwards and internal injury might well result. It is to minimize this possibility that the shoulder strap idea has been introduced.

The safety belt under review is known as the R.K.N., and is composed of grey-blue Terylene material which has been tested to withstand a load of one ton, in fact a vehicle of this weight has been suspended by the belt alone. The lap strap has a central lock which may be released with one finger and this, of course, is essential if the safety strap is to cause no inconvenience to a person entering or leaving the seat.

The R.K.N. safety belt produced by Delaney Gallay Ltd.

The whole efficiency of the safety strap lies, as previously stated, in the attachment to the car, and it would obviously be undesirable to rely solely on the seat fixture. The attachment is, therefore, made to a $\frac{3}{8}$ in. loop bolt that is passed through the steel floor with load spreading plates above and below the floor. Where a timber floor is encountered the loop bolt is passed through to secure a fixture to a convenient chassis member, although some difficulty might be experienced here in the case of bodies using a flexible mounting. No doubt in such a case further stiffness could be introduced into the floor itself.

The point of attachment to the floor itself varies according to the type of seat under consideration, and the method that causes greatest inconvenience in the rear compartment is the one to be adopted in the case of seats having tilt backs. For the normal rigid-backed seat, the attachment encroaches to only a small degree on the rear floor area, and there is a third method which offers no encroachment at all. Now that interest has been shown in the use of safety straps for cars exported to the United States of America, there seems little doubt that this safety precaution will be fitted in increasing numbers on cars used in this country.

FURNACES FOR THE ALUMINIUM INDUSTRY

DURING 1957 The Morgan Crucible Company Limited, Battersea Church Road, London, S.W.11, introduced a new electric die-casting furnace and a new oil-fired, basin tilting furnace after a considerable period of research. The die-casting furnace was developed in collaboration with Birlec Limited of Birmingham. It is a crucible furnace, electrically heated, of 300 lb charge capacity for maintaining aluminium and aluminium alloys, and combines the proved advantages of crucible melting with the undoubted benefit of electric heating.

The electric die-casting furnace has fully automatic temperature control, which keeps the metal accurately at the specified casting temperature, thereby relieving the operator of all furnace duties, so that he can concentrate on die-casting. Automatic control of electric heating ensures constant performance from the furnace and complete freedom from variations in metal quality. If an automatic time switch is fitted, the furnace can be ready for use at the start of the shift, without requiring an operator to switch it on before the normal working time.

The basin tilting furnace is complementary to the die-casting furnace, since it is a bulk melting furnace for feeding maintaining furnaces. It has all the qualities necessary for a bulk melter; a high melting rate, simple operation and lip pour. These qualities, coupled with crucible melting, give highest quality metal with minimum metal loss. A special feature of this basin tilter is the built-in tilting mechanism which gives a smooth lip pour with finger-tip control. The resulting steady stream of metal allows the ladles to be kept in one position throughout the pour.

DIESEL ENGINE PERFORMANCE

*A Technique for Reducing Smoke and Improving Output**

M. Alperstein, W. B. Swim, P. H. Schweitzer

FUEL economy is the outstanding advantage of the diesel engine. This derives from two facts; the compression ratio of the diesel is roughly double that of the automobile petrol engine, and the diesel can burn a lean mixture that a spark is unable to ignite. In most other respects, however, the diesel is inferior to the Otto engine. In the latter the fuel and air are mixed *before* they enter the cylinder and most of the fuel is in the vapour phase at the time of ignition. Therefore a rather rich mixture can be completely burned without thermal decomposition; hence carburettor engines generally do not smoke. Secondly, the spark plug is able to ignite fuels of low ignition quality, which in a compression-ignition engine would either ignite too late or not at all.

The technique described in this report represents a little step away from the diesel engine in the direction of the Otto engine. It is a step that preserves all the diesel advantages and alleviates two disadvantages, incomplete mixing of air and fuel and late combustion caused by long ignition lag. Broadly, the method consists in introducing part of the fuel in a fine mist or fume into the intake manifold while the remainder is introduced in the conventional manner through the high pressure injection system into the cylinder near top dead centre position of the piston.

Manifold introduction of fuels into a compression-ignition engine was experimented with at Pennsylvania State University as early as 1941, and is reported in the master thesis of C. W. Overbeke, "*Manifold Introduction of Hydrocarbons as an Aid for Starting*," *August*, 1942. Various means of vaporization and carburation were used for introducing auxiliary fuels such as hexane, heptane, five different diesel fuels, white petrol, hydrogen peroxide, benzoyl, methyl alcohol, benzene, cetane and diethyl ether. Ether, cetane, hexane and low boiling point diesel fuels were found to be the most effective; acetone, benzene and ethyl alcohol were least effective or were detrimental when introduced into the intake manifold. It was pointed out then that slow oxidation and pre-flame reactions were believed to be responsible for the effect on ignition of the manifold introduced fuels.

McLaughlin, Pinotti and Sigworth[1] used manifold introduction of fuels—mostly petrol and liquid petroleum gas—to kill smoke and/or boost power, as was also done by Derry[2] and others and by Lyn[3] in England. In all these researches the main fuel was injected into the diesel cylinder in the conventional manner and a smaller amount of auxiliary fuel was introduced into the intake manifold as an aid to combustion. Havemann[4] and others, on the other hand, carburetted alcohol as a main fuel and injected a small amount of diesel oil and various vegetable oils to ignite the compressed air-alcohol vapour mixture. In this manner they successfully burned alcohol, which otherwise could not be ignited by compression ignition. A similar scheme was used during the war by the German air force. They developed the "R Fluid," a less volatile ether, which they used in place of a spark plug. Ignition was produced by the injection of "R Fluid," before top centre, into the compressed air-petrol vapour mixture. Still earlier, a pilot spray of diesel fuel was used in the same manner in duel-fuel gas engines.

In the last three instances the main fuel was inducted through the intake manifold and a relatively small amount of auxiliary fuel injected. But ignition was always released by the injected fuel. In fact, without the injection of a *liquid* jet, the gaseous main fuel would not ignite at all, except at a very high compression ratio.

Most recently, Arnold[5] and others used supplementary fuels of high ignition quality to help burn low ignition quality fuels, which otherwise would burn only sluggishly or not at all. Previous investigators used injection nozzles or carburettors to introduce supplementary fuel into the intake air. Such devices give rather coarse sprays with a mean drop size of 20 microns or more. For the investigations on which this paper is based, an apparatus called Micro-Fog was employed. It produces a very fine mist consisting of droplets under 4 microns. This gave amazingly good results. Subsequently, the investigations were extended to cover a

**A report on work carried out in the Department of Engineering Research, Pennsylvania State University.*

TABLE I. FUEL PROPERTIES

Fuel	Chemical constitution	Spec. gravity (60/60) deg F	Kin. viscosity cS at 100 deg F	Boiling point deg F	Octane number	Cetane number	Latent heat of evaporation B.Th.U/lb	High heating value B.Th.U/lb	Low heating value B.Th.U/lb	Stoichiometric air:fuel ratio
N-hexane	C_6H_{14}	0·659	0·41	155	26		156	20,900	19,432	15·2
N-heptane	C_7H_{16}	0·684	0·52	209	0		155	20,850	19,408	15·1
Ethyl alcohol	C_2H_5OH	0·785	1·11	172	99		367	12,780	11,604	9·0
Diethyl ether	$(C_2H_5)_2O$	0·714	0·28	94·3			165	15,840	14,560	11·1
Amyl nitrate	$C_5H_{11}NO_3$	1·000	0·80	297			150	10,600	9,810	7·48
Iso-propyl nitrate	$(CH_3)_2CHNO_2$	1·058	1·82	216			137	8,210	7,580	5·55
90 octane primary Ref. fuel blend	90% C_8H_{18} 10% C_7H_{16}	0·701	0·58	211	90		155	20,600	19,184	15·1
Diesel fuel, WCN	H/C=0·154	0·850	2·7	680 E.P.		43·6		19,610	18,260	14·5
Diesel fuel, CCD	H/C=0·156	0·836	2·75	623 E.P.		52·4		19,780	18,430	14·5
JP-5	H/C=0·134	0·883	1·27	504 E.P.		22·9		19,380	18,075	14·3

TABLE II.
STANDARD CONDITIONS FOR CFR ENGINE TESTS

3¼ in bore, 4½ in stroke, 900 r.p.m.	17 : 1 compression ratio
Lubricating oil	S.A.E. 30
Lubricating oil temp. deg F	135±1
Lubricating oil pressure	25-40 lb/in²g, depending upon speed
Water jacket temperature, deg F	210±1
Injection angles	6 deg B.T.C. for 600 r.p.m.
	7-13 deg B.T.C. for 900 r.p.m., depending upon the test
	15 deg B.T.C. for 1,200 r.p.m.
Inlet air temperature, deg F	100±2*
Inlet air pressure	Atmospheric±0·3 in Hg†
Exhaust gas pressure	1·0±0·5 in water

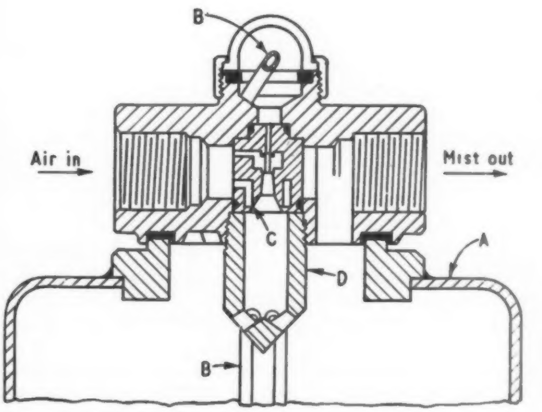
**Inlet air temperature was above 100 deg F in the tests run with the vaporizer unit, and the actual value for temperature depended upon the fuel used and the rate of introduction.*

†A series of tests was run under supercharged conditions in which the inlet pressure was maintained at 5 in Hg gauge.

TABLE III.
STANDARD CONDITIONS FOR PETTER AVA-1 ENGINE

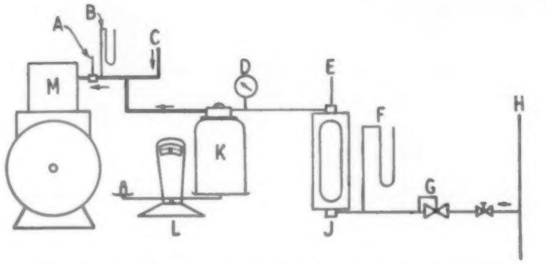
Lubricating oil	S.A.E. 30*
Injection angles, not adjustable	24 deg B.T.C. up to 1,500 r.p.m.
	28 deg B.T.C. 1,500 to 1,800 r.p.m.
Inlet air temperature	Room temperature
Inlet air pressure	Atmospheric —0·6 in H_2O at 900 r.p.m. —0·9 in H_2O at 1,200 r.p.m.

**S.A.E. 10 oil was used for the non-firing tests.*



A fuel tank; **B** fuel riser; **C** atomizer holes, four; **D** diffusion plug

Fig. 1. Above: Cross section of Norgen Micro-Fog



A inlet temperature; **B** inlet pressure; **C** air intake system; **D** pressure gauge; **E** air temperature; **F** air pressure; **G** regulator; **H** compressed air service line; **J** Rotameter; **K** Micro-Fog unit; **L** sensitive scale; **M** CFR engine

Fig. 2. Micro-Fog supplementary fuel system

variety of auxiliary fuels and several methods of introduction. The properties of the fuels used are given in Table I.

Test equipment

Most of the tests were performed on a single-cylinder CFR cetane test engine, but some were carried out on a single-cylinder Petter AVA-1 diesel engine. The CFR engine has a Ricardo Comet type swirl chamber head and a fixed compression ratio of 17 : 1. The Petter engine was of the open chamber type with a compression ratio of 16·5 : 1. It was air cooled. Tables II and III give the standard operating conditions for the two engines.

Supplementary fuel was introduced by four different methods: a mist generator, a commercial pneumatic spray nozzle, a carburettor, and vaporizer. The mist generator was an experimental model of the commercial Micro-Fog lubricator made by C. A. Norgren Co. A cross-section of the device is shown in Fig. 1. In operation, liquid passes through air jets issuing from four atomizer holes. The so atomized liquid is then carried by the air through a series of separator

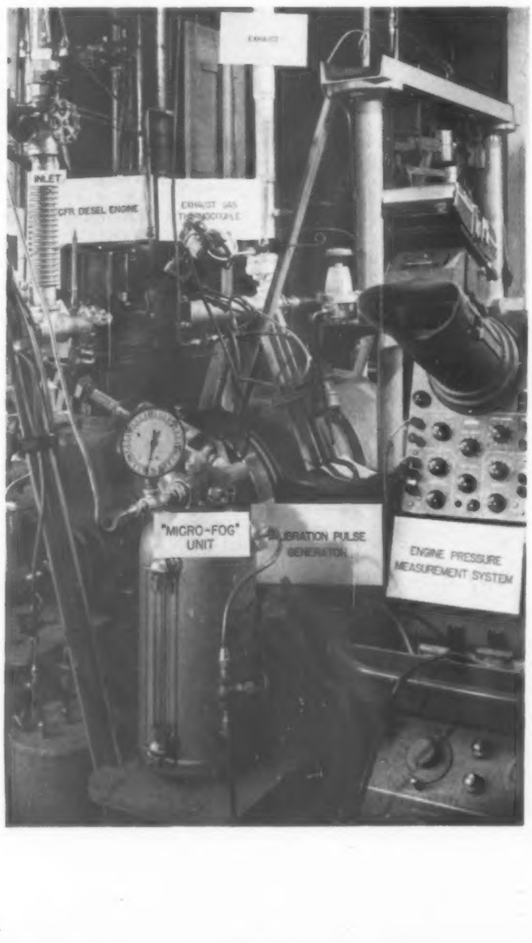
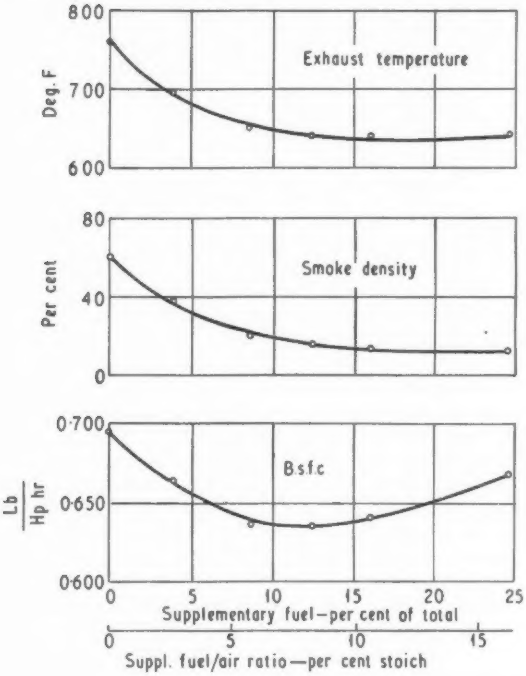
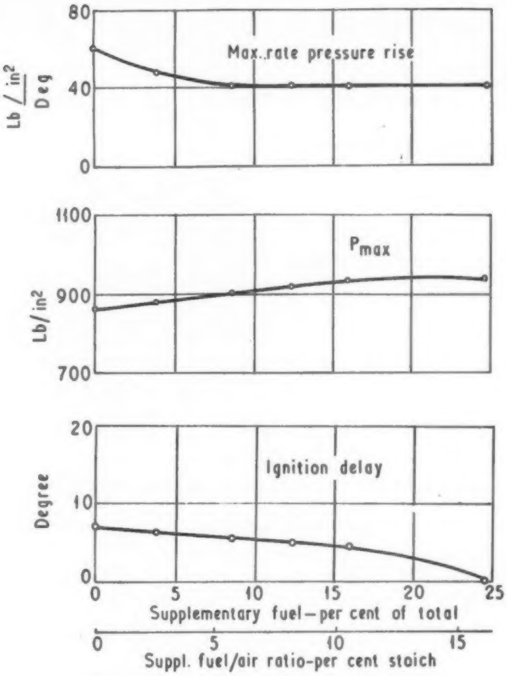


Fig. 3. Right: Micro-Fog installation on CFR engine





B.m.e.p. 57·7 lb/in². Both fuels 44 C.N. diesel fuel. Supplementary fuel introduced by Micro-Fog. CFR engine, 900 r.p.m. Injection begins at 13 deg B.T.C.

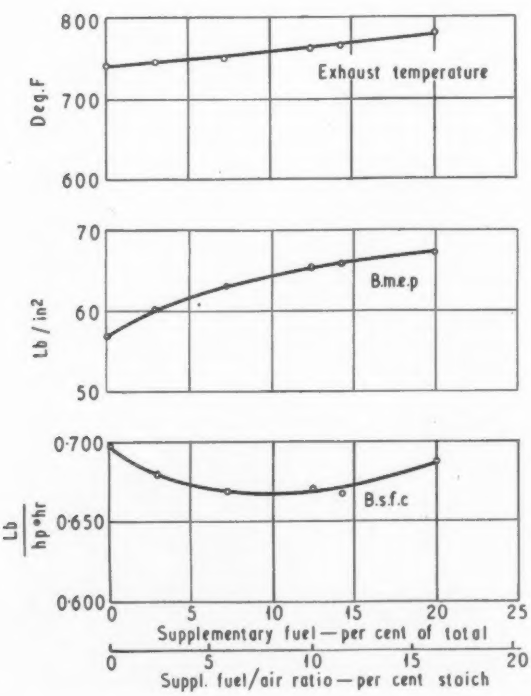
Fig. 4. Effect of supplementary fuel on engine performance at constant brake load

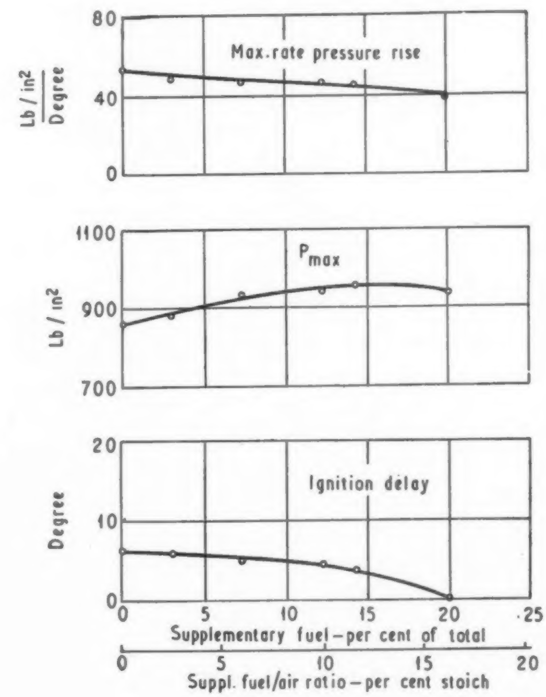
Fig. 5. Effect on ignition lag and rate of pressure rise at constant brake load

Fig. 6. Effect on exhaust temperature, power and fuel consumption at constant smoke density, 60 per cent

Fig. 7. Effects on pressure and ignition lag at constant smoke density, 60 per cent

Both fuels 44 C.N. diesel fuel. Supplementary fuel introduced by Micro-Fog. CFR engine, 900 r.p.m. Injection begins at 13 deg B.T.C.





passages where the heavier droplets are removed. The remaining mist is then conveyed by the air stream with controlled pressure into the manifold. This mist, the drop sizes are all under 4 microns, is so fine that it does not wet a hand or a glass surface held in front of it.

The pneumatic atomizing nozzle used in the test programme was made by Spraying Systems Co. The fuel rate was controlled by means of a needle valve feeding to a constant head fuel reservoir. Air at pressures ranging from 10-15 lb/in^2 passed through an annulus around the fuel nozzle and impinged upon the fuel, thus atomizing it. Air flow requirements were less than 1/40th those of the Micro-Fog, but the mass median droplet size was 59·5 microns, a rather coarse spray.

The carburettor was adapted from a Briggs and Statton 2½ h.p. engine carburettor. It comprised a venturi, a metering orifice and a constant head fuel reservoir. Fuel was introduced into the fuel reservoir from a fuel tank by a needle valve controlling the supplementary fuel rate. The carburettor was mounted in a by-pass of the primary air supply line. In operation, a constant percentage of the engine air requirements passed through the carburettor. The atomized fuel, carried by this by-passed air, joined the primary air supply and was then admitted to the engine. To ensure adequate air-flow through the carburettor, throttling of the primary air line was necessary.

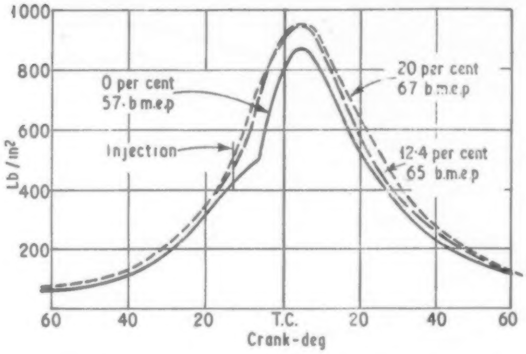
The vaporizer consisted of a metal container through which all the air to the engine was fed. Fuel was fed to the vaporizer from a gravity feed tank controlled by a needle valve. The bottom of the container was electrically heated above the boiling point of the supplementary fuel so that the fuel would flash immediately upon hitting it. This resulted in high intake air temperatures, depending upon the supplementary fuel used.

Instrumentation for the tests included: strain gauge type and piezo-electric pressure pick-ups and an oscilloscope with camera to record pressure-time traces, a stroboscopic timer, flowraters to measure primary and supplementary air flow, an electric air heater, exhaust gas and inlet air thermocouples, CRC and Von Brand smoke meters, and a General Radio noise level meter. The majority of the tests were conducted with either the smoke density or the brake load held constant. A diagrammatic sketch of the test set-up with the CFR engine and the Micro-Fog is shown in Fig. 2, and the actual set-up is shown in Fig. 3.

Tests with normal fuels

The results of constant brake load tests are shown in Figs. 4 and 5. A good grade of 44 C.N. fuel was used for both the main and supplementary fuels. The Micro-Fog device was used for introducing the supplementary fuel. It can be seen from Fig. 4 that the addition of the supplementary fuel caused a reduction in smoke density from 60 per cent (full load starting point) to 12 per cent. A significant decrease in the fuel consumption, 8·6 per cent at 11 per cent supplementary fuel, was also noted, with a marked decrease in exhaust gas temperature. Fig. 5 shows that the ignition lag drops to zero at 24 per cent supplementary fuel rate; then pre-ignition occurs. Smoother, quieter operation was obtained owing to the lower rate of pressure rise, although the maximum pressure increased.

Figs. 6 and 7 show the results when the smoke density was kept constant at 60 per cent CRC smoke meter reading, which is barely visible exhaust. The b.m.e.p. was found to increase about 18·6 per cent when 20 per cent of the total fuel was supplied with the intake air, and the fuel consumption was found to decrease only about 4·3 per cent when the supplementary fuel was approximately 10 per cent of the total. Firing pressures and rates of pressure rise change as in the constant brake load tests, but ignition delay becomes zero at 20 per cent supplementary fuel rate.



44 C.N. diesel fuel used for both main and supplementary fuels. CFR engine, 900 r.p.m. Injection begins at 13 deg B.T.C.

Fig. 8. Pressure diagrams for 0, 12·4 and 20 per cent supplementary fuel at constant smoke density

Superimposed pressure diagrams for normal injection and for two rates of supplementary fuel are shown in Fig. 8. The higher compression pressure when supplementary fuel is used indicates the presence of exothermic pre-flame reactions. It shortens the ignition lag and decreases the maximum rate of pressure rise.

The tests illustrated in Figs. 4-8 were performed with an injection advance of 13 deg B.T.C., which was the optimum injection angle for standard running. From the pressure traces, it appeared that later injection would further improve the engine performance. Tests with variable injection angles have indeed shown that specific fuel consumption decreased with retarding injection until the optimum was reached at 7 deg advance. The specific fuel consumption was then 9·8 per cent better with 14·5 per cent supplementary fuel than with normal operation.

In a conventional engine a slight advance of injection usually makes the engine rougher but improves the fuel consumption; on the other hand, when supplementary fuel is used the opposite result is obtained. This is shown by the following test results:—

Injection advance	Brake specific fuel consumption lb/b.h.p-hr	
	Normal operation	**10½ per cent supp. fuel**
13 deg	0·635	0·624
7 deg	0·655	0·591

The smoke density was kept constant at 60 per cent during the tests. In Fig. 9 test results at 600, 900 and 1,200 r.p.m. are shown. At higher speeds the gain is greater.

Test results for three methods of introducing the supplementary fuel are shown in Fig. 10. These were constant smoke density tests and each of the three methods showed an improvement in performance. The greatest improvement was effected by using the mist generator. The vaporizer unit was penalized by the higher inlet air temperature necessary to operate the unit (250 deg F compared with 100 deg F). This was inherent in the operation of the vaporizer, since the plate used to vaporize the fuel also heated the inlet air. In order to avoid separation of the fuel from the air stream, there was no provision for cooling the inlet air prior to induction.

Fig. 11 shows superimposed indicator diagrams of normal and supplementary fuel cycles with the three different methods of introducing the supplementary fuel. It will be noted that for the greater part of the compression stroke the pressure increases for the vaporizer and Micro-Fog are

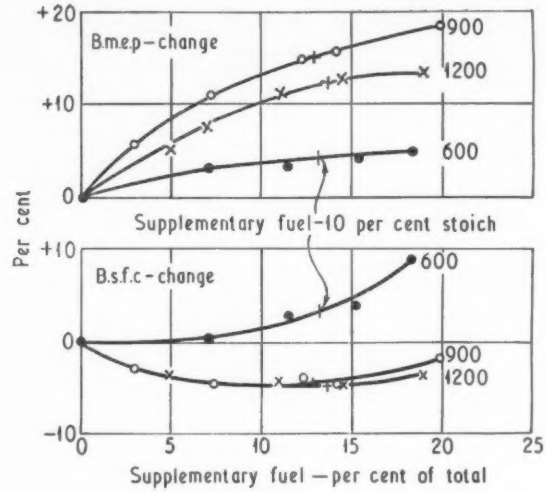


Fig. 9 The effect of engine speed on supplementary fuel addition at constant smoke density, 60 per cent

practically equal, but the Micro-Fog becomes more effective during ignition and combustion. In comparison with the vaporizer, the pneumatic spray nozzle produces less pressure increase before ignition but greater increase after ignition.

Tests were run using five different supplementary fuels; (n-heptane, n-hexane, alcohol, petrol and diesel fuel) in combination with various primary fuels and using three introduction methods. When diesel fuel was used as the supplementary, the mist generator gave the best results. As more volatile supplementary fuels are used, the method of introduction becomes less critical. Figs. 12 and 13 show the results of a series of constant brake load tests using the five different supplementary fuels introduced by the mist generator while injecting a high grade diesel fuel. The introduction of diesel fuel brought the greatest improvement. This is also true if the tests are run at the maximum load corresponding to 60 per cent smoke (CRC). Fig. 14 shows superimposed indicator diagrams for the five supplementary fuels.

Tests were conducted on the Petter AVA-1 engine to determine the effect in an open chamber diesel engine. The pneumatic spray nozzle was used to introduce the supplementary fuel. The engine had a constant geometric timing of 24 deg B.T.D.C., and it was found that engine performance with neat 52 cetane fuel left little room for improvement. Constant heavy load tests gave, as shown in Fig. 15, smoke density and fuel consumption reductions of 33 per cent and 4 per cent respectively. With a low cetane, 31 C.N. injected fuel, the engine showed greater reductions in smoke density and fuel consumption (43 per cent and 10 per cent respectively) see Fig. 16.

The tests already mentioned were run with zero inch gauge pressure in the inlet manifold. By using shop air the inlet manifold pressure was raised to 5 in Hg gauge, and under these conditions indicator diagrams such as shown in Fig. 17 were obtained. They show that with supercharge the optimum amount of supplementary fuel is less than with natural aspiration. The optimum was obtained with 8·7 supplementary fuel, when the b.m.e.p. was 89·2 lb/in² and the specific fuel consumption was 0·581 lb/b.h.p-hr, improvements of 9·4 per cent and 5·4 per cent respectively.

Sub-standard fuels

Long ignition lag is the reason for the unsatisfactory combustion of low cetane fuels. Since the technique used in these tests shortens the ignition lag, it was to be expected that it would facilitate the combustion of low cetane fuels. A 90 octane petrol, a mixture of 90 per cent iso-octane and 10 per cent n-heptane, was tried first. With only normal injection, the engine refused to run at low load and was erratic at high load. When approximately 20 per cent of the same fuel was introduced by Micro-Fog the engine ran steadily. Smoother operation was obtained by the introduction of 12 per cent of 44 diesel fuel into the intake manifold; this reduced the ignition lag from 20 deg to 10 deg C.abs. and the rate of pressure rise from 90 to 20 lb/in²-deg. The injection advance was 9 deg B.T.C. in these tests. Even so the operation was far from satisfactory and the fuel consumption excessive (0·73 lb/b.h.p.-hr).

The next primary fuel tested was a JP-5A fuel of 23 cetane number. With normal injection only, operation was unsatisfactory at light load and was impossible at idle. When straight amyl nitrate was used as supplementary fuel introduced by a carburettor at the rate of approximately 1/800 lb per lb of air there was a definite improvement. Steady and smooth operation was obtained down to idle, although the fuel consumption was high. N-propyl nitrate and iso-propyl nitrate gave similar results.

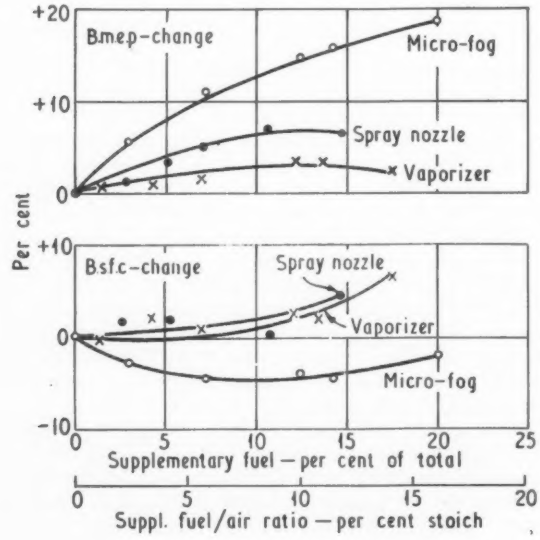


Fig. 10. Left: Effects of various methods of introducing supplementary fuel at constant smoke density, 60 per cent

Fig. 11. Below: Pressure diagrams for three different methods of introducing supplementary fuel at constant smoke density, 60 per cent 44 C.N. diesel fuel used for both main and supplementary fuels

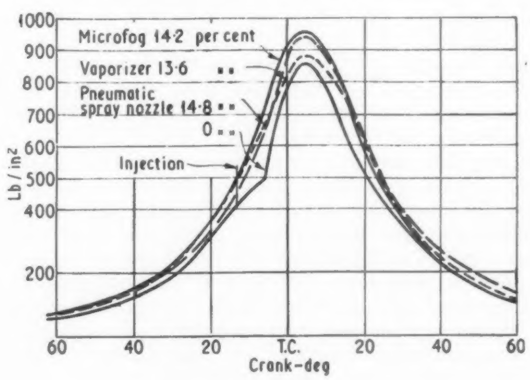


TABLE IV.

MINIMUM RATES OF SUPPLEMENTARY FUELS FOR STABLE OPERATION ON 29 C.N. MAIN FUEL; SUPPLEMENTARY FUEL; NEAT 52 CETANE FUEL

	Approximate minimum rate, per cent
Neat 52 cetane fuel	25
97·5 per cent 52 cetane and 2·5 per cent amyl nitrate	23
95 per cent 52 cetane and 5 per cent amyl nitrate	18
90 per cent 52 cetane and 10 per cent amyl nitrate	14

A constant amount of iso-propyl nitrate carburetted into the intake manifold gave the results shown in Fig. 18 with JP-5A primary fuel in the CFR engine. For this diagram, the total fuel consumption was plotted and the lower heating value of iso-propyl nitrate disregarded. Even alpha methyl naphthalene (C.N. $=0$) could be burned with this scheme. High cetane diesel fuel introduced by a pneumatic spray nozzle gave similarly good results.

Low load operation with low cetane fuel was still more difficult with the Petter engine. However, the engine could be made to operate satisfactorily on low cetane fuel if the proper supplementary fuel was used. Tests revealed that the type of supplementary fuel had a marked effect on engine operation, and that the higher the cetane number of the supplementary fuel the smaller the amount required for acceptable performance. Table IV shows the minimum rates of supplementary fuel, for several different types, required to obtain steady combustion when running on 29 C.N. fuel. Excessive amounts (upwards of 50 per cent) of low cetane supplementary fuel were required to achieve stable operation, and even then the performance was relatively poor.

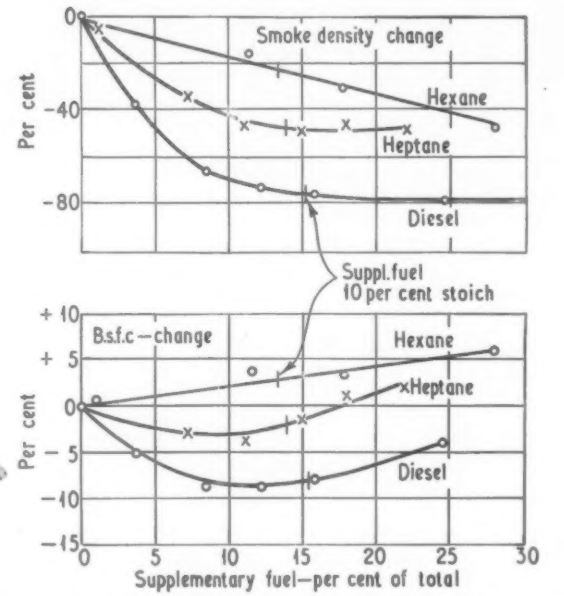
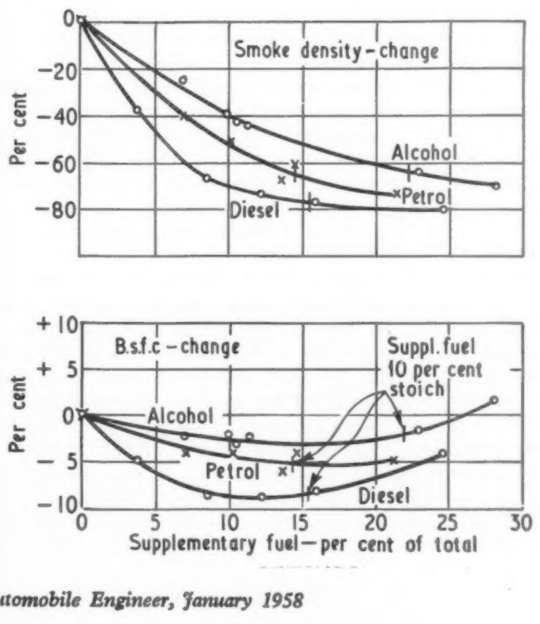


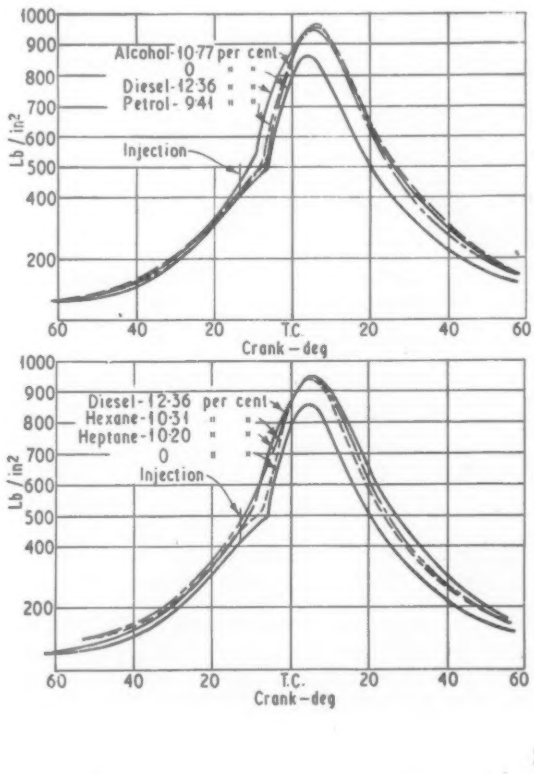
Fig. 12. The effects of hexane, heptane and diesel supplementary fuels on engine performance at constant brake load, b.m.e.p. 57·7 lb/in²

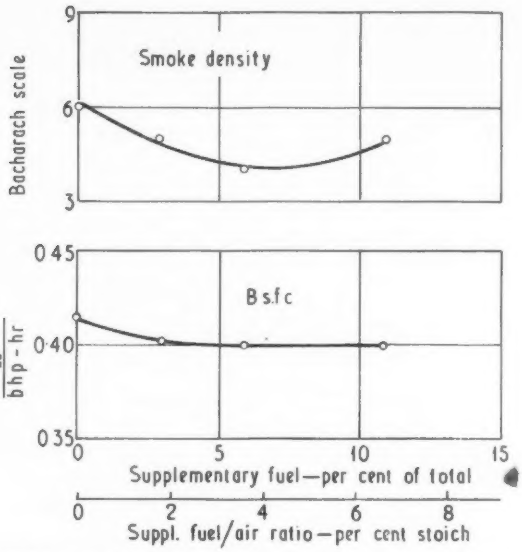
It was thought that the addition of low viscosity supplementary fuel might result in poor lubrication of the cylinder walls, and because of this piston blow-by was watched continuously and an accurate record kept of the engine wear. The CFR engine has been run, using supplementary fuel, approximately 375 hours between overhauls. The blow-by did not increase appreciably during the course of the test programme, not did measurement of rings, cylinder and piston made during the overhaul period reveal any abnormal wear.

Fig. 14. Right: Pressure diagrams for various supplementary fuels on smoke density, 60 per cent. Primary fuel, diesel 44 C.N.

Fig. 13. Below: The effects of alcohol, petrol and diesel supplementary fuels on engine performance at constant brake load, b.m.e.p. 57·7 lb/in²

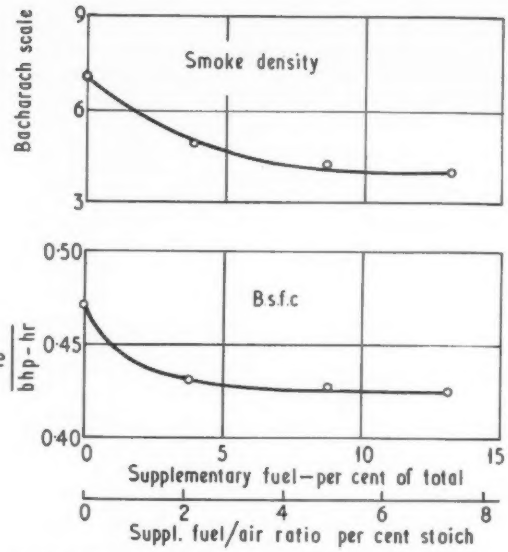






B.m.e.p. 97 lb/in². Both fuels 52 cetane. Supplementary fuel introduced by spray nozzle

Fig. 15. The effect of high cetane diesel supplementary fuel on smoke density and fuel consumption at constant brake load



Both fuels 31 cetane. Supplementary fuel introduced by spray nozzle

Fig. 16. The effect of low cetane diesel supplementary fuel on smoke density and fuel consumption at constant brake load, 86 lb/in²

How does it work?

It has been shown that Micro-Fog introduction of ordinary diesel fuel, in an amount approximately 15 per cent of the total, increased smoke-limited power output and also increased thermal efficiency. It can be imagined to help diesel combustion in two ways: better air utilization due to pre-mixing and more complete combustion due to the chemical effect. Of the earlier investigators, McLaughlin[1], Derry[2], Havemann[4] attributed the improvements mainly to better air utilization, while Lyn[3] emphasized the beneficial effects of pre-flame reactions on diesel combustion.

To analyze the factors contributing to improved engine performance on supplementary fuel and to determine the relative effect of each, Fig. 19 was prepared, using data from a constant smoke test in which both the fuel and air rates were measured. The increase in b.m.e.p. may have resulted from (a) the ability of the engine to burn more pounds of fuel per pound of air for the same degree of smoke density, better air utilization; (b) an increase in the thermal efficiency of the engine as a result of decreased ignition lag and faster combustion; (c) a combination of both. It was certainly not caused by the supercharging effect of the supplementary air because pressure and temperature were strictly kept at their standard values, atm $\pm 0{\cdot}3$ in Hg, 100 deg ± 2 deg F, for these tests except when a vaporizer was used.

With an excess air factor of λ and a stoichiometric air:fuel ratio of 14·5 the theoretical b.m.e.p. of the engine is:

$$\text{b.m.e.p.} = 180\,\frac{\text{brake thermal eff.}}{0{\cdot}325}\;\frac{\text{volume eff.}}{\lambda}$$

With x per cent supplementary fuel, that portion of the fuel needs no excess air ($\lambda=1$), therefore the increased b.m.e.p. will be:

$$\text{b.m.e.p.}' = (1-x)\,180\,\frac{\text{brake thermal eff.}}{0{\cdot}325}\;\frac{\text{volume eff.}}{\lambda} + x\,180\,\frac{\text{brake thermal eff.}}{0{\cdot}325}\;\frac{\text{volume eff.}}{1}$$

From this the relative boost figures are:

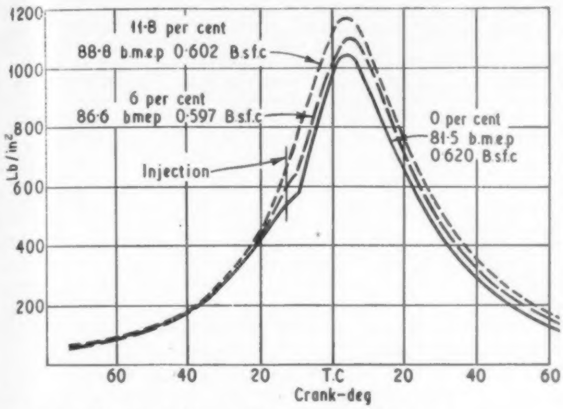
$$\frac{\text{b.e.m.p.}' - \text{b.m.e.p.}}{\text{b.m.e.p.}} = x(\lambda - 1)$$

Under standard conditions the b.m.e.p. was 56·8, the brake thermal efficiency was 0·1856, the volumetric efficiency was 0·85, and the excess air factor was 1·54; therefore the relative boost would be $0{\cdot}54x$ or roughly one half of the percentage of supplementary fuel. The b.m.e.p. plot shows two curves, one the experimental curve and the other computed using a constant brake thermal efficiency. From this it can be seen that both factors contribute to the improved performance.

The investigations on which this paper are based show that both factors contribute to improved performance. In general it is correct to say that if power boost came from better air utilization alone, it would amount to less than the percentage of supplementary fuel, whereas the power increase was generally greater than the percentage of supplementary fuel. Improved combustion efficiency must have played a part. This becomes obvious from the great differences in the effects of supplementary fuels of different chemical compositions.

Pre-flame reactions can improve combustion in at least two ways. First, by shortening the ignition lag; secondly, by helping combustion after ignition has set in. Fig. 20 shows the effect on ignition lag of introducing, with Micro-Fog and with a vaporizer, increasing amounts of alcohol, 90 octane petrol, hexane, heptane and 44 cetane diesel fuel as supplementary fuels. It will be noted that the effect of alcohol and high octane petrol on the ignition lag is almost zero, while that of hexane, heptane and diesel fuel is very pronounced. But the effect on performance was not in the same order. As can be seen from Fig. 12, hexane reduced the smoke the least amount and actually increased the fuel consumption. On the other hand, alcohol and high octane petrol were far from without effect on combustion. They, then, must act through a mechanism other than shortening the ignition lag.

The manifold introduced fuel obviously exerts an influence on combustion in several ways. First, during the relatively long period of compression it is subject to slow oxidation, which, being an exothermic process, must raise the pressure and temperature of the cylinder charge more than if pure air were compressed. For this to happen, the hydrocarbon and oxygen molecules must be in contact, which can effectively occur only if the hydrocarbon is in gaseous state. Volatile fuels could be expected to excel in this effect. In this slow process of oxidation, however, few if any of the C molecules combine directly with O_2 molecules into CO_2 and the H_2 molecules with O molecules into H_2O. Combination takes place rather through the intermediate combustion products such as peroxides and aldehydes. The heat



Both fuels 44 C.N. diesel. Inlet manifold pressure 5 in Hg gauge

Fig. 17. Pressure diagram for a supercharged engine at constant smoke density, 60 per cent, and 0, 6·0 and 11·8 per cent supplementary fuel

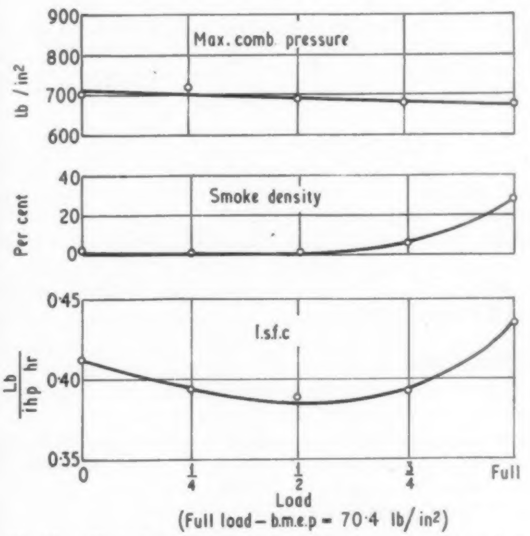


Fig. 18. The effect of using iso-propyl nitrate as supplementary fuel in conjunction with a low cetane, 23, main fuel

generated in such reactions is less, but these intermediate combustion products react more rapidly with oxygen than the original hydrocarbon. Therefore, slow oxidation may shorten the chemical as well as the physical portion of the ignition lag.

Parallel to the oxidation process another process is going on, that of pyrolysis or thermal decomposition, commonly called cracking. Under the influence of heat and pressure, the hydrocarbon molecule breaks into smaller molecules. This process is endothermic. It takes 80 k cal/mole to break a C–C bond, 100 to break a C–H bond, 150 to break a double bond C=C, and 200 to break a triple bond C≡C. For this reason such thermal decomposition may never take place if conditions of easy oxidation are present. Thermal decomposition can, however, take place in the absence of oxygen. The hydrocarbon need not be vaporized to crack.

Thermal cracking has been identified[7,8], as being responsible for soot deposits and exhaust smoke. But from the standpoint of ignition quality, hydrocarbons that crack easily like long chain paraffins are the best diesel fuels, and those that crack hard like ring chain aromatics are the worst. No matter how undesirable cracking is after ignition, it is probably very helpful, perhaps essential, before ignition. R. O. King[9] has put forth the theory that ignition generally is initiated by solid nuclei like carbon flakes. If the combustion air is free from such nuclei, a thermal decomposition of the fuel is needed which produces such carbon particles that can act as nuclei to initiate combustion. If the influence of supplementary fuel on combustion is through early oxidation, the volatile fuels of few carbon atoms should be best. If the effect is by speeding up ignition through thermal

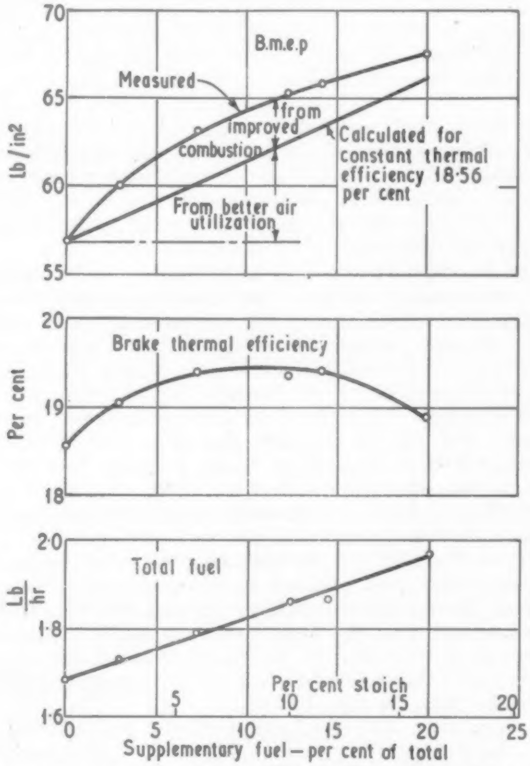


Fig. 19. Source of power increase with supplementary fuel introduced by the Micro-Fog unit

Fig. 20. Ignition lag against rate of supplementary fuel at constant smoke density, 60 per cent

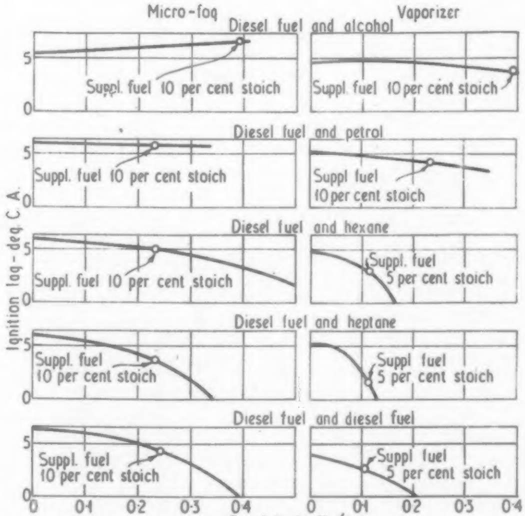


TABLE V. NON-FIRING TEST RESULTS—CFR ENGINE

	Fuel: Air ratio lb/lb	Per cent Stoichiometric	I.m.e.p. lb/in^2	Exhaust temp. deg F	Max. cylinder pressure, lb/in^2
Air alone	0	0	0	120	520
Ethyl alcohol	0·01197	10·8	0·8	120	520
90 octane primary fuel blend	0·00938	14·2	1·7	127	525
44 cetane diesel fuel	0·01072	15·5	7·6	137	565
N-heptane	0·00863	13·1	6·4	150	570
N-hexane	0·01129	17·2	22·0	185	640

No main injection, fuel introduced by Micro-Fog, speed 900 r.p.m., 100 deg F inlet air temperature, 3 zero Hg gauge inlet air pressure

decomposition, the high cetane fuels containing many carbon atoms should be best.

Non-firing tests

In order to gain a better insight into the mechanism of pre-flame reactions, tests were conducted with a non-firing engine. Injection was completely eliminated, and fuel was introduced only into the intake manifold of a CFR engine motored at 900 r.p.m.

With some fuels auto-ignition occurred when the amount of supplementary fuel exceeded a certain quantity, others did not auto-ignite at all. Fig. 21 shows pressure traces with 44 cetane diesel fuel, 90 octane petrol and alcohol, while Fig. 22 shows the same with heptane and hexane. On each, the solid line shows the air compression-expansion line without fuel introduced as a very fine mist. On the hexane and heptane traces the dashed line shows supplementary fuel in the amount which gave the best power output on previous tests with diesel fuel injection, the dotted lines show where incipient ignition set in, and the dash-dotted lines show fully developed spontaneous ignition.

Table V shows the tabulated observations on maximum cylinder pressures, exhaust temperatures and computed i.m.e.p. It shows hexane as being the most effective, alcohol the least effective, with diesel fuel in the middle. This is in marked contradiction to the supplementary fuel tests, which showed diesel fuel as the best and hexane the poorest of those tried, see Figs. 12 and 13.

A significant difference between the two tests was that in the non-firing tests there was no ignition. The shortening of the ignition lag, therefore, could have no part in the process. What obviously played a decisive part was the heat generated during compression owing to slow oxidation of the manifold-introduced fuel. This caused an amazing rise of pressure and temperature and an appreciable indicated power output, even though most of the heat introduction occurred during the compression stroke.

Inasmuch as in these tests the fuel was introduced in small liquid droplets by Micro-Fog, and oxidation on any appreciable scale can only take place in gaseous state, the fuel had to vaporize before it got oxidized. Therefore, volatility must have played an important part in the process. This would explain the good showing of hexane but fails to explain the poor showing of alcohol. On the other hand, the presence of thermal decomposition during the compression would result in the creation of lighter molecules, some of which would oxidize at a faster rate than the original heavier molecules.

If this speculation is sound, the thermally unstable high cetane fuels should again assume top positions if the volatility factor is eliminated. The first non-firing tests were performed with Micro-Fog and were then repeated with a vaporizer. In this latter case the fuel was in the vapour state when it entered the cylinder and fuel volatility could no longer have an effect. Fig. 23 shows the indicator diagrams obtained under such conditions.

Applications

The study of pre-flame reactions brought closer understanding of auto-ignition. A true understanding of the mechanism through which ignition accelerators work could easily lead to clearer concepts of combustion in both spark and compression ignition engines, including the role of knock suppressing agents. What occurs in a fraction of a

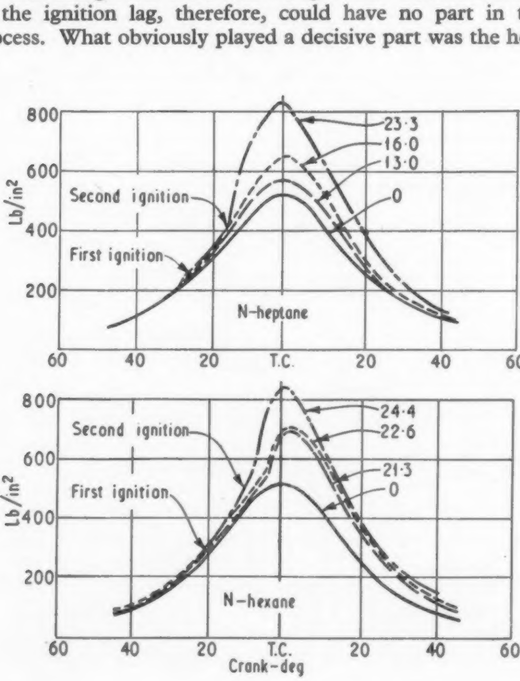
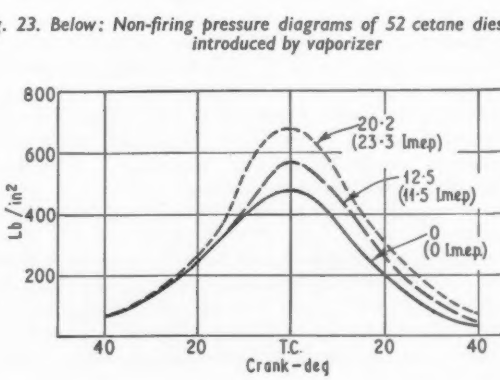


Fig. 22. Left: Non-firing pressure diagrams for n-heptane and n-hexane

Fig. 23. Below: Non-firing pressure diagrams of 52 cetane diesel fuel introduced by vaporizer



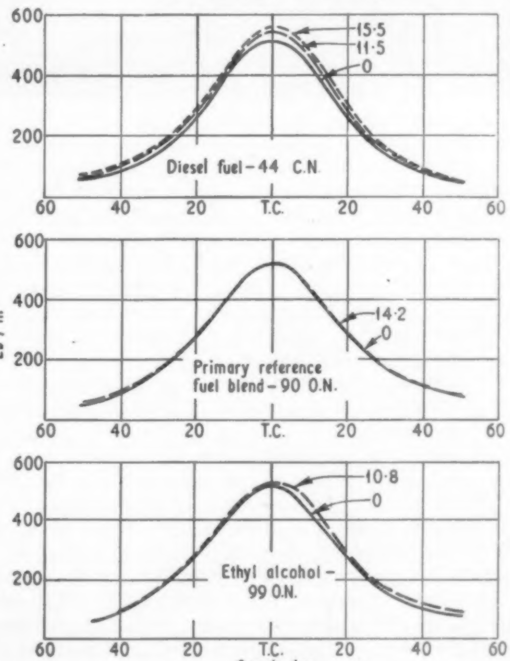
milli-second during normal combustion can, by such technique, be stretched out and studied in steps.

The most obvious use is for reduction of exhaust smoke or increase of smoke-limited power output. Both for convenience and effectiveness it is recommended that the same normal diesel fuel should be used as primary and supplementary fuel. The complication of a two-fuel system is thereby avoided. It is true that a low ignition quality fuel such as petrol, used in quantities of 30 to 40 per cent of the total fuel rate could bring higher power boosts,[1,2] but intake pipe induction of diesel fuel in such quantities would cause pre-ignition under certain operating conditions and control over the engine would be lost. When the amount of supplementary fuel is restricted to 10-15 per cent of the total, regular diesel fuels are superior to petrol, provided a fine spray is used in both cases.

Apparently, judicious use of the technique would improve the fuel consumption on all types of diesel engines. For best fuel economy it is essential that a mist generator or equivalent is used. If Micro-Fog is used, full-load fuel economy could be improved some 10 per cent; with an ordinary pneumatic spray nozzle the saving will be about 5 per cent.

At present three factors militate against the use of the Norgren Micro-Fog mist generator on commercial engines: cost, bulk, and air consumption. The largest capacity Micro-Fog, the experimental model shown in Fig. 3, has a capacity of 0·2 lb/hr with an air flow rate of 0·4 lb/b.h.p-hr. At a ratio of 15 per cent, 0·06 lb of supplementary fuel is needed per hour, therefore the device shown is barely adequate for 3½ h.p. An air pressure of 18 lb/in²g is needed to produce this flow which requires 0·26 theoretical horse-power or approximately 0·35 actual horse-power. This means that all the fuel saving would be spent to drive the air compressor to supply the air needed to operate the Micro-Fog. Furthermore, the present Micro-Fog is very bulky. The Micro-Fog is so large and takes so much horse-power because only about 2 per cent of the total amount of atomized liquid is discharged as a mist with drop sizes under 4 microns; the other 98 per cent with larger drop sizes is returned to the tank.

Fig. 21. Non-firing pressure diagrams. The numbers indicate percentage stoichiometric fuel : air ratio



A less extravagant atomizer is the ordinary pneumatic spray nozzle, which also operates with compressed air but uses only about 1/50th as much air per pound of liquid as the Micro-Fog. A 1,000 h.p. engine burns approximately 400 lb of fuel per hour at full load of which approximately 60 lb/hr would be supplementary fuel that would take about 120 pounds of air per hour. The compression of that quantity of air to 50 lb/in²g would absorb about 3·3 h.p., or 0·33 per cent of the engine output.

This technique also makes it possible to use fuels of low ignition quality in compression-ignition engines. High speed diesel engines have difficulty in burning diesel fuels of about 35 cetane number at light load or idle, although operation may be satisfactory at heavier loads. In such cases, the introduction of part of the main fuel as a mist will allow satisfactory operation at all loads.

If the main fuel has still lower ignition quality, for example, motor petrol of approximately 20 C.N., the use of the main fuel as the supplementary will help, but better results are obtained by using a higher centane fuel doped with a cetane improving additive. The ignition quality of the injected fuel ceases to limit its use in a diesel engine if the supplementary fuel is properly selected. For example, 29 cetane diesel fuel can be satisfactorily used in a small high-speed open chamber engine with a supplementary fuel of 44 C.N. doped with 2½ per cent amyl nitrate.

Diesel engined trucks afford a good field for applying this technique. These vehicles are normally equipped with an air compressor for furnishing compressed air for braking. This equipment should be adequate to supply air to a pneumatic spray nozzle mounted in the intake manifold. A tee in the main fuel line could conduct fuel to the spray nozzle, and the air flow could be controlled by the fuel rack of the injection pump in such a manner that it would begin to operate at around 7/8 load and both air flow and fuel flow would increase with the rack setting up to an acceptable overload. This would enable diesel trucks to climb hills without smoking. Neither the fuel bill nor the maintenance costs would be increased.

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LATHAM SUPERCHARGER

An Axial Flow Unit Designed for Quantity Production Economically with Conventional Tools

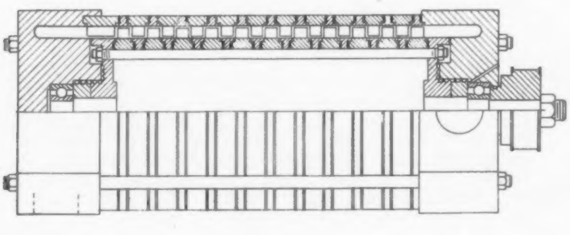
ONE of the reasons why superchargers have not been more widely accepted for private cars is that most of them are noisy. The Latham Manufacturing Co., of West Palm Beach, Florida, United States, claim that this objection has been overcome with their axial flow compressor. Hitherto, the development of axial flow compressors has not received as much attention as that of the radial flow type. This is because radial flow superchargers are more suitable for aircraft engines, particularly of the reciprocating piston type, of the radial cylinder-layout, and therefore vast sums of money have been spent by governments on their development for military applications.

The following arguments are advanced by the Latham Manufacturing Co. in support of the axial flow compressor for automobile applications. Although centrifugal compressors are well suited to aircraft engine installations, where the compressor housing and diffuser can be of large diameter, in automobile applications space limitations are such that the diameter must be restricted, and therefore efficiency suffers and speeds have to be high. Reliability tends to be impaired by these high speeds, and acceleration is inevitably sluggish. The need for rapidly accelerating the supercharger drive components sets a number of design problems tending to demand complex drive arrangements. One of the reasons why the radial flow type supercharger is so satisfactory for aircraft applications is that the range of speeds over which these engines are operated for most of their running time is not as great as in automobile applications.

Roots type compressors have been used perhaps more than any others on automobile engines. Their principal disadvantages are the noise they generate and the restricted pressure ratio obtained. Vane type compressors are not so widely favoured, because of the relatively large number of components rubbing against one another and wearing.

The Latham axial flow compressor has been developed over the last ten years. In producing this design, the manufacturers have sought to fulfil the principal requirements as follows. The unit has been intended, from the outset, for production in large quantities by conventional tools. A range of sizes is available and cost has been kept to a minimum by virtue of an improved method of blade fabrication and attachment. The design is such that a wide range of surge-free operation is obtained. Installation is relatively simple because the unit is of uniform cross section. It is also claimed that with this supercharger, the fuel consumption, over the entire speed range, is improved by an average of 2 m.p.g., as compared with the standard unblown engine. Another marked advantage of the unit is that it has neither gears and similar moving components subject to wear, nor interlocking rotors that must be accurately synchronized. Its single rotor is carried on two Norman-Hoffmann ball bearings, lubricated through grease nipples.

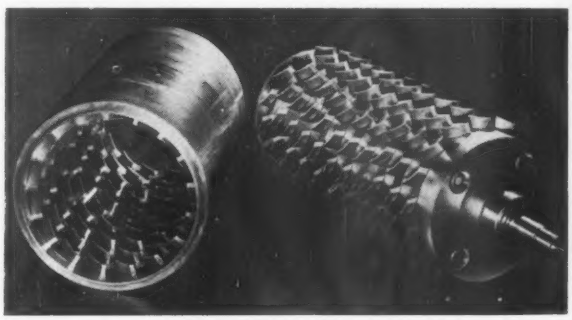
Fig. 1. All the components, including the shafts and tie bolts, of the Latham supercharger are of aluminium alloy. This obviates difficulties with regard to differential rates of thermal expansion



In this supercharger, the rotor and stator are composed of a series of interlocking blade rings and spacer rings, each of similar cross section and diameter, as shown in Fig. 1. The manufacturers state in their patent specification, that the entire compressor, including the shafts and tie bolts, is made of the same metal, preferably aluminium, to avoid differential rates of thermal expansion. To obtain an annular air passage of constant cross section throughout the length of the compressor, the blade pitch is varied from stage to stage. This can be done without increasing the cost of the blades. A half-tone illustration of the rotor and stator blade assemblies is shown in Fig. 2.

Fabrication and attachment of the blades are effected as follows. They are punched from aluminium alloy sheet of high tensile strength. All the dimensions, except the thickness, of the blanks thus formed are in excess of the finished requirements. These blades are pressed into slots milled in the blade carrier rings. The width of the slots is nominally the same as the thickness of the blades, but the curvature varies from stage to stage. On the other hand, the curvature of the blades as produced by the die is that of the slots in the centre stage; thus, at all the other stages, the blades have to be sprung into their slots.

Fig. 2. All the blades on both the rotor and stator assemblies are identical until they reach the manufacturing stage at which final machining of each separate blade and carrier ring assembly is done



After the over-size blades have been pressed home, each blade ring and the end faces of its blades are together turned to the finished dimensions. As can be seen from Fig. 3, the finished ring and blade assembly of each stage is of shouldered cross section; and is spigoted into the adjacent T-section spacer rings, the arms of the T-sections fitting over the shoulders of the blades and of the blade rings, which they thus retain. Accurate alignment of the blades is ensured by metal-to-metal abutment between them and the spacer rings. Interposed between the flanks of the spacer rings and the blade rings are gaskets, which are compressed a predetermined amount by the tie bolts that hold the whole

assembly together. The method of attachment of the blades is similar for both the stator and the rotor.

Non-surge operation is obtained over a wide range by virtue of the cross section of the leading edges of the blades. When the flow is considerably restricted, the air tends to approach the blades in the direction of arrow B in Fig. 3, and is therefore approximately in alignment with the leading edge profile. On the other hand, when the compressor is running under its normal operating conditions, the air tends to approach the blades in the direction of arrow A and is therefore approximately in alignment with the blade itself.

For most passenger car installations, a Carter YH side-draught carburettor is mounted on each side of the super-charger inlet volute, which is at the front. The carburettor throttles are interconnected by a link that passes underneath the stator case. At the rear, the discharge volute is attached, by means of an adaptor, to the standard air intake manifold of the engine to which the supercharger is fitted.

The eleven-stage version of this compressor, which is sold at a retail price of less than 300 dollars, has 363 blades manufactured to a tolerance of 0·001 in. Its outside diameter, over the compressor section, is 6½ in and the distance across the flats of the volutes is 6¾ in. Thus, so far as height is

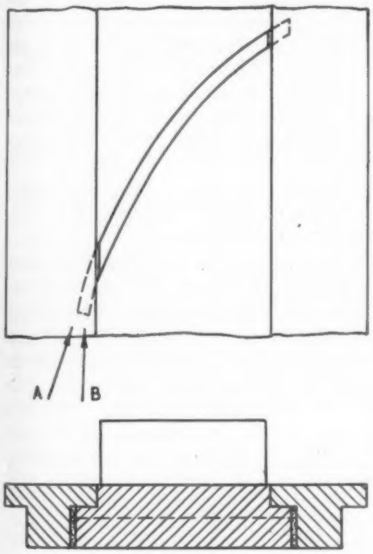


Fig. 3. Detail view showing how the blades are retained in their carrier rings by T-section spacer rings

concerned, the unit compares favourably with the carburettor and air cleaner installations on the majority of American V-eight engines currently in production.

A neat installation of this supercharger on the Ford Mercury L-head engine is shown in Fig. 4. It would appear that the length of the supercharger is not too great for its accommodation above engines of this type. A close-up view of another installation can be seen in Fig. 5.

The manufacturers claim that the most important property of this type of compressor is its high efficiency. Whereas comparable superchargers of other types absorb in the order of 30 b.h.p. at maximum output, delivering at a manifold boost pressure of approximately 3 lb/in², it is stated that under these conditions the Latham unit absorbs only 10 b.h.p. and the manifold boost pressure obtained is 8 lb/in². Because of this, the supercharger can be driven by a 1 in wide, flat belt.

This method of driving the supercharger is claimed to be as effective as it is simple. A large diameter pulley is bolted to the crankshaft pulley. The supercharger drive pulley, on the other hand, is small in diameter. Had a V-belt been employed, the pulley diameters would, of course, have been too large to be practicable for most automobile applications.

A Gilmer belt is employed. It is made up of fine braided

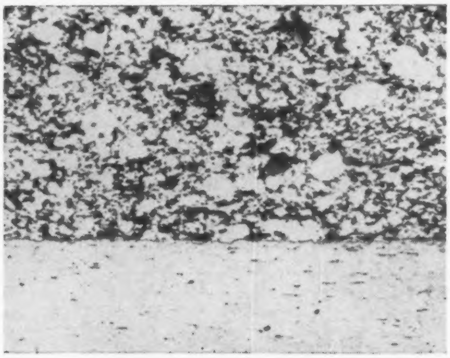
Fig. 4. A Ford Mercury L-head engine fitted with a Latham supercharger used in conjunction with two Carter YH sidedraught carburettors. The throttles are interconnected so that they operate together

steel wire cables, bonded in turn to a textured nylon facing. A spring-loaded idler pulley is mounted between the drive and the driven pulleys to tension the belt. This belt does not stretch perceptibly, because of its steel reinforcement. Its nylon friction facings are said to be extremely wear-resistant. Another advantage is that since it is only 0·070 in thick, it is very flexible and permits a good degree of wrap-round to be obtained on the small diameter pulley.

Tests have been carried out on a 1956 Ford, overhead valve, V-eight engine, equipped with this supercharger. This engine has a relatively low compression ratio, of 8 : 1, and therefore is well suited for these tests. The only modification, apart from fitting the supercharger conversion kit to the engine, was an increase in the cylinder diameter to give a running clearance of 0·005 in between the piston and the cylinder. The fuel : air ratio was adjusted to give maximum output.

First the engine was run in its naturally-aspirated condition. In this test, a maximum of 163 b.h.p. was developed at 4,400 r.p.m., and the maximum torque was 235 lb-ft at 3,200 r.p.m. For the second test, the complete Latham supercharger assembly was installed on the engine. This test gave a maximum of 210 b.h.p. at 5,000 r.p.m., and the maximum torque was 282 lb-ft at 3,000 r.p.m. At the speed at which maximum power was developed, the manifold pressure was 6½ lb/in². The increase in horse power obtained by supercharging, as compared with the naturally-aspirated condition, was as much as 72 per cent at 2,000 r.p.m.

Fig. 5. Adaptors, such as those shown in this illustration, can be provided for connecting the discharge volute to the standard air intake manifold of almost any engine currently in production



Photomicrograph of plate section showing the bond between the bronze facing and the steel support plate

SINTERED BRONZE-FACED CLUTCH PLATES

Production of Components for the Borg-Warner Automatic Transmission at Sintered Products Ltd.

A FEATURE of the modern automatic transmission unit is the compact, multi-plate clutch running in oil. As more makes of cars are being equipped with automatic transmissions and as more vehicles so equipped are going into service, the numbers of clutch plates required will rapidly increase. Already the figures are substantial and should the use of automatic transmissions become more or less general, as in the U.S.A., the quantities will assume significant proportions. An indication of this was given in a recent article* on barrel finishing, in which was described an automatic plant operated by one American concern for processing 3,200 clutch plates per hour.

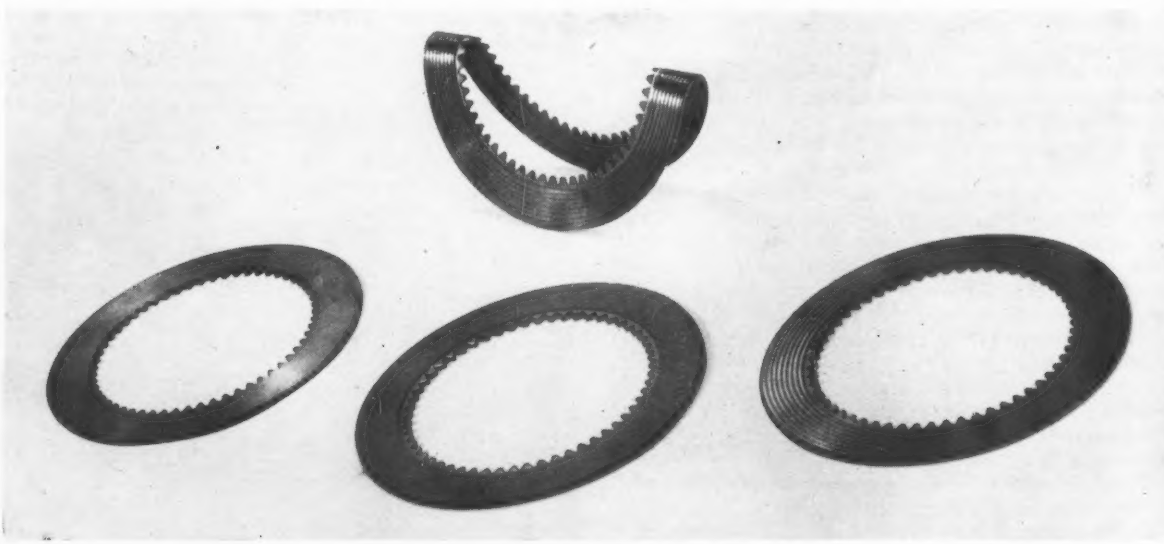
For the automatic transmission unit manufactured by Borg-Warner Ltd., Letchworth, Herts., the clutch plates are produced, faced, and finished for Borg-Warner Ltd. by Sintered Products Ltd., Hamilton Road, Sutton-in-Ashfield, Notts. This company manufactures a number of grades of sintered bronze or iron friction material, variously suitable for a wide range of both wet and dry applications. These materials, designated S1, S2, S3, *et seq*, are marketed under the trade name of "Don" by Small and Parkes Ltd., Hendham Vale Works, Manchester 9. The composition and the operating characteristics of each of four of these materials, together with the method of production, have been fully described in this journal†.

The material used for the Borg-Warner plates is the S3 grade, which is specifically intended for wet applications where high static and dynamic friction coefficients are required. Under static conditions the coefficient is 0·13 to 0·15 at a pressure of 20 lb/in^2. Curves plotting the dynamic coefficients, determined in a test run in oil at various application pressures and rubbing speeds, are given in the graph. As regards composition the main constituent is copper, but small percentages of lead, graphite, silica, iron, and other ingredients are included.

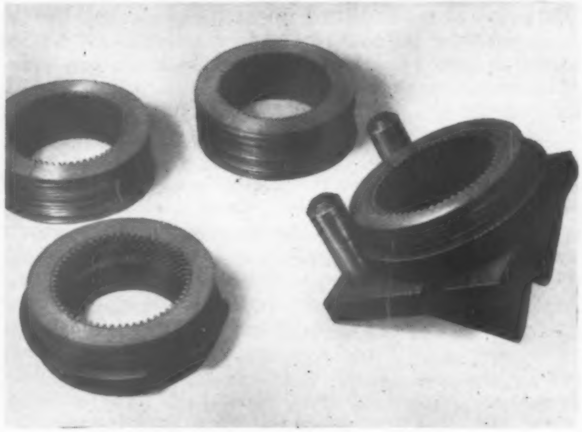
On a clutch incorporated in an automatic transmission there is little scope for such practices as "bedding down" or "wearing in", and consequently the plates must be manufactured to what may be regarded as relatively close tolerances and a comprehensive inspection system be employed to ensure conformity in the finished component. The facing rings are produced individually on a hydraulic compacting press at a pressure of nominally 12·5 ton/in^2 to a thickness of 0·22-0·24 in. Obviously, a ring approximately 5$\frac{3}{16}$ in external diameter and 3$\frac{13}{16}$ in internal diameter of such a thin section requires very careful handling while in the "green" state. After the stripping of the dies and ejection, the compact is slid onto a flat sheet metal plate and on this

*"*Automobile Engineer*", *October* 1957
†"*Automobile Engineer*", *July* 1955

Copper-plated steel support plate, the composite plate as it leaves the sintering furnace, and the machined and ground finished plate. A test specimen folded through 180 deg illustrates the effectiveness of the bond between the friction rings and the support plate



Compacted facing ring ejected from the press tools



Assembly of components on stacking fixture prior to sintering

is removed from the press table and stacked on a tray ready for assembly.

Fully annealed steel to En 32 or S3 specification is used for the steel support plates, with En 6, En 8, or En 12 steels as permissible alternatives in the event of supply difficulties. The sheet material is $0{\cdot}33 \pm 0{\cdot}002$ in thick and the surfaces must be free of rolling scale. It is pierced and blanked, with 0·004 in and 0·005 in tolerance on the internal and external diameters respectively. The bore of the plate is blanked to form 57 teeth of 16 diametral pitch, the pitch circle diameter being 3·5625 in. Check dimension for these teeth is 3·478-3·4835 in between 0·090 in diameter pins.

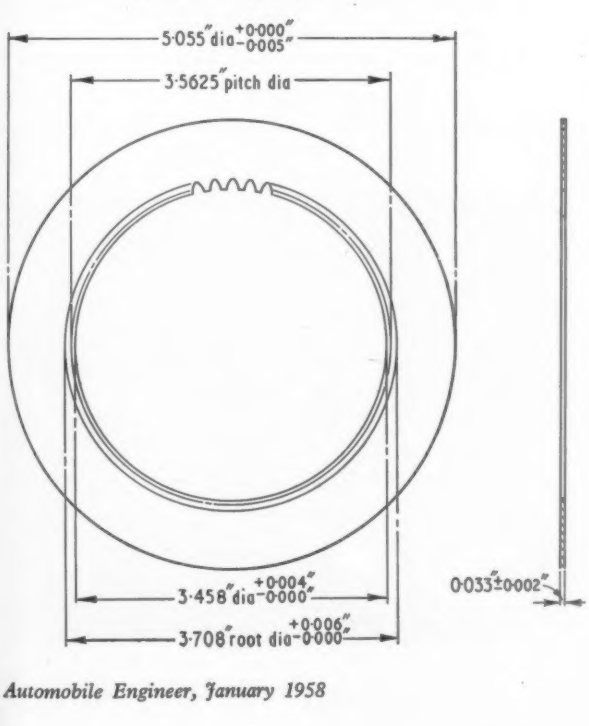
All burrs are removed and the plate must be parallel within 0·002 in and flat within 0·005 in. After chemical cleaning the plates are then copper plated. The copper coating protects the surfaces of the plate from oxidation during sintering and also facilitates the formation of a metallurgical bond between the plate and the friction material.

A photomicrograph of a sectioned plate, giving some indication of the intimate character of the bond between the sintered bronze facing ring and the steel support plate, appears at the head of this article. On the same page, the effectiveness of the bond is shown in the photograph of a test plate bent through 180 deg. Although the facing rings are locally cracked there is no evidence of lifting and no particles are detached.

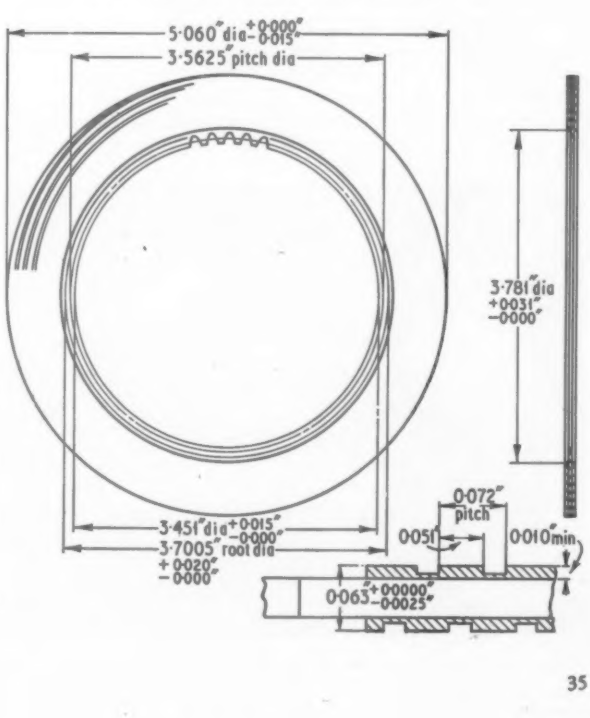
Support plates and compacted facing rings are assembled in stacks on an inclined plane fixture, the component items aligning concentrically by the engagement of their peripheries with two perpendicular pins. Individual assemblies are separated by a carbonaceous compound to ensure parting after sintering.

From ten to twenty assemblies are stacked on a ring, of graphite or of "Thermex" heat-resisting steel, which provides a convenient method of handling and serves as a spacer when they are loaded in the sintering furnace. These furnaces are of the bell type and are electrically heated. Multiple stacks of assembled parts on their spacer rings are built up on the furnace base, with a total of 400 to 500 assemblies constituting a charge. Pressure to a value between 10 and

Copper-plated support plate in En 32 steel



Finish ground friction plate, faced with Don S3 material



250 lb/in² on the projected ring areas of the six stacks is applied to keep the plates flat and also to ensure the friction material rings bond satisfactorily to their support plate. This pressure is provided by the weight of the furnace bell, or by an air pressure cylinder operating through a lever and a central bar.

The assemblies are sintered in a protective atmosphere, confined in an atmosphere can that encloses the charge and is sealed at the base. Over this is placed the external furnace bell with the heating elements and the refractories mounted in the walls. To sinter, the charge is brought up to the appropriate temperature of 760-800 deg C and held at that value for 30 minutes. The overall time cycle for this operation, including loading and unloading the furnace, is between 3 and 6 hours, determined according to the weight of the charge. A furnace unit comprises a heating bell and a plurality of bases and atmosphere cans. The heating bell is removed at the termination of the critical temperature period and transferred to another base already loaded and ready for operation. During the cooling phase the atmosphere can remains in position until the temperature of the charge has fallen to a value precluding the possibility of oxidation.

During the sintering process the combination of heat and compressive pressure occasions some slight displacement of the material of the component parts. Allowance is made for this effect, of course, in the dimensions of the parts as initially produced. From the drawings of the support plate and the composite plate it will be noted that the nominal internal diameter, over the addenda of the teeth, is reduced from 3·458 in to 3·451 in. The tooth measurement over 0·090 in diameter pins is also reduced from 3·478-3·4835 in to 3·4738-3·481 in.

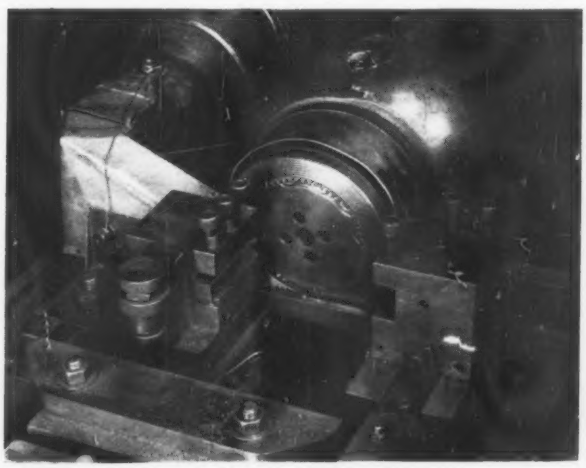
The composite rings are machined on both friction surfaces and on the outside diameter. This operation is performed on a cam automatic; the plate being set up on the profiled spigot of a magnetic face plate on the spindle. A front form tool mounted on the main slide faces the friction surface, removing about 0·005 in, and simultaneously cuts in it the scroll-type oil groove. In section the groove is rectangular, 0·021 in wide and 0·006 to 0·009 in deep, and is formed with sharp corners. It is machined as a single-start spiral, running from the inside diameter to the outside diameter, with a pitch of 0·072 in and leaving a land 0·051 in wide. The surface is continuously brushed to clean the grooves as they are turned and, since the facing material contains finely divided lead and silica, dust extraction equipment is fitted. A tool mounted on the back slide machines the outside diameter.

Assembled plates stacked on the base of the sintering furnace

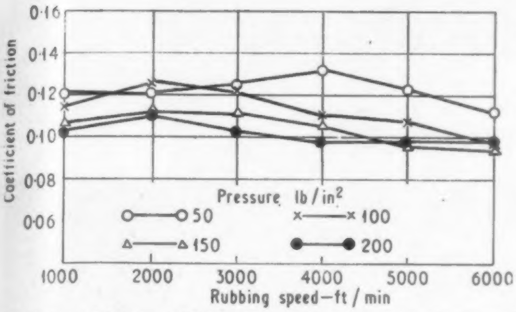
On completion, the plate is unloaded, reversed, replaced on the face plate, and the operation is repeated on the other face. The feed stroke of the form tool is automatically increased to compensate for the reduction in thickness of the work by the first operation and to size the work. When both facings are machined the overall thickness is 0·68 in and the minimum acceptable thickness of either facing is 0·010 in. No change is made in the hand of the scrolling when the plate is reversed, so when assembled in the clutch unit the scrolls on the contiguous surfaces of adjacent plates are of opposite hand. Since the grooves serve mainly as escape routes for oil when the clutch is engaged, this feature is of no operational consequence.

After machining, the plates are ground on each surface

Set-up for facing friction ring, machining scroll oil groove and turning outside diameter



Finish grinding the plates on machine equipped with 6-station indexing table



Coefficient of friction of S3 friction material at different pressures and rubbing speeds

to reduce the overall thickness from 0·068 in to 0·063-0·0605 in. The operation is performed on a horizontal grinding machine with a similar set-up to that used for finishing piston rings. Plates are loaded onto and unloaded from a 6-station rotatable table which is automatically indexed by an air-operated ratchet mechanism. As a plate reaches the grinding station it is transferred to a rotating magnetic face plate. The grinding head then strokes; the wheel moving radially across the friction facing and then returning. Production rate exceeds 400 plates per hour. The operation is repeated with the plate reversed to finish the other facing.

Inspection is arranged to provide a comprehensive check on accuracy and a practical guarantee of consistency in performance. There is a 100 per cent visual examination and a check on parallelism. A 20 per cent check is made on outside diameter; inside diameter, over pins; thickness; flatness, by passing through spaced plane surfaces; and the tooth circular space, by an expanding rig and a dial gauge. The depth of the scroll groove is also checked on 10 per cent of the plates.

The Numericord Data-Control System for Machine Tools

IN 1952 Giddings and Lewis Machine Tool Co., Fond du Lac, Wisconsin, U.S.A., sponsored a research at the Massachusetts Institute of Technology for the design and development of a director that could produce control information to operate a multi-axis machine tool system. The data-control system developed as a result of this work is known as the Numericord system. This system has four distinct operational phases: the preparation of the mathematical data; the checking and transcription of this data on to punched paper tape; the relating of this data to time, including the carrying out of linear interpolation; and finally automatic operation of the machine tool.

Preparation of the mathematical data is usually referred to as programming. It is the only technical function involved. The programme engineer receives a drawing of the part to be machined, decides what cutters are to be used, what fixtures are necessary, and specifies all other related machining conditions such as feed rates. He also decides the sequence in which the individual faces of the part shall be machined; this is specified by entering on a programme sheet the co-ordinates of points through which the cutter is to pass. The arithmetical calculations entailed can be effected on an electronic digital-computer or, for simpler shapes, on a standard electric desk calculator. The completed programme sheet is passed to a copy-typist who operates the keyboard of a transcription unit. This unit checks the logic of the information on the sheet, and if the logic is correct, transcribes it as a series of punched holes in a paper tape.

The director

The punched paper tape is used to enter data into the director, the next stage of the preparation system. Data are entered on the tape in blocks, and each block contains the incremental displacements required of the machine slides along each axis, together with the time over which this motion is to take place. The director then produces synchronized control-information that ensures the cutter will move along a straight line between its initial point and a new point implied by the displacement values quoted for each axis. Straight-line interpolation is not limited to particular planes bounded by the principal axes of the machine tool. It can also be carried out in skew planes where x, y and z incremental displacements are given. In the director, the control information is also related to time to give the correct cutter feed rate. The continuous control-information so derived is then recorded on magnetic tape.

Magnetic tape

The magnetic tape is 1 in wide and 0·04 in thick. There are 14 recording tracks spaced across its width. The allocation of these tracks may differ from machine to machine, depending upon the application, but certain specific functions are controlled by signals recorded on specific tracks. For identification, the tracks are numbered from 1 to 14, numbering starting from one edge and continuing in sequence across the width.

On one of the middle tracks (track 7) a 200 cyc/sec reference signal is recorded. On the five tracks adjacent to it (tracks 5, 6, 8, 9, 10) are recorded square-wave signals for controlling the motion of five machine-tool slides. These tracks are allocated in this manner so that should the magnetic tape be slightly misaligned under the reading heads—slightly askew for example—phase accuracy between the important signals that control slide movements will scarcely be affected. Tracks 4 and 11 are used for controlling the magnetic tape transport mechanism.

Signals that operate on-off auxiliary functions such as supply of coolant, control of spindle drive and operation of clamps are recorded on the six remaining tracks. The control information operating auxiliary functions is impressed on the magnetic tape as sinusoidal signals of either 200 cyc/sec or, 4,000 cyc/sec frequency. The appropriate auxiliary function will be switched on and will remain on as long as the particular signal-frequency that is used to operate it is impressed on the tape. For example, the coolant will be supplied as soon as a 200 cyc/sec signal is detected on track 1, and will remain on as long as the signal is recorded on this particular track. A 4,000 cyc/sec signal on track 1 will, however, initiate operation of a side clamp.

Therefore, although only six tracks are provided for auxiliary functions, 12 auxiliary functions can be controlled by allocating a different frequency and track to each. Two signals of different frequencies can be recorded simultaneously on the same track, and can be separated for identification by high-frequency and low-frequency filters in the control circuit. Track 4 has the playback stop for stopping the machine tool completely, while track 11 has planned stops for stopping the machine at the operator's discretion. The Numericord system and applications are described in full detail in the January, 1958, issue of our associated journal *Aircraft Production*, in one of a series of articles based on information collected by the author during a visit made specifically to investigate the latest practices and probable developments in the United States of America.

Balancing Engines

A Ford Development in America

BALANCING engines dynamically is not a new process in the U.S.A. Lincoln and Mercury engines for example have been balanced in this way since 1954 by using large, stationary, electrically driven machines. This method, however, required handling all engines regardless of whether they needed balancing or not. With statistical quality control, the majority of engines do not require balancing, but it has been necessary to check them all on special machines to find out which are in tolerance and which are out. As a result, considerable time has been lost.

In a new process, developed by the Ford Motor Company, which combines balancing with hot testing so that it becomes part of the regular cycle, this unnecessary handling is eliminated. The percentage of engines that need balancing at this point is still only a fraction of the total so the majority are checked in balance while being hot tested and are then passed on. The new process not only saves time but reduces floor space requirements. In addition the equipment has a production capability of ¼ oz-in with a linear response of ⅛ oz-in throughout the range. This is especially advantageous when weight slugs are used. Another advantage is that without external couplings to drive the engine there is no external source of unbalance added to the engines under test.

In the Ford plant, a conveyor automatically delivers engines to the hot test area where each stand is equipped with an electronic balancer built by International Research and Development Corp., of Columbus, Ohio. The principal components of a balancer are two vibration pick-ups, a console containing amplifiers, filters and associated electronic circuitry, two meters for indicating the amount of unbalance, and a stroboscopic unit that indicates both engine speed and location of unbalance.

The two vibration pick-ups are the sensing elements for vibration caused by unbalance and are mounted on an arm attached to the conveyor carrier. One pick-up points radially to the front and the other radially to the rear correction plane of the engine. The front correction plane is on the V-belt pulley and the rear correction plane on the flywheel. Alternating currents generated by the pick-ups respond to the mechanical motion of the engine and are proportional to the vibration amplitude and r.p.m. of the engine. The phase of the electrical output bears an established relationship to the physical location of unbalance.

Output from the pick-ups is fed to the console which houses the electrical equipment that operates the system. In addition to the dual channel circuitry for simultaneous treatment of signals from both the front and rear pick-ups, the console contains a dual calibration system controlled by one master selector switch to accommodate different types of engines. Each electronic channel contains filters, amplifiers, integrating circuits, and plane separation networks. The purpose of the filters is to reject all unwanted vibration signals—explosion pulses, reciprocating pulses, vibration from outside the engine, etc.—except vibrations resulting from the engine's rotational speed of 1,500 r.p.m. The plane separation network eliminates the effect of unbalance at one end of the engine upon the reading at the opposite end.

Signals from the console are fed to two meters mounted above each test. These meters simultaneously present the amount of unbalance at the front and rear of the engine. In addition to direct calibration in ounce-inches, each meter is graduated into four major divisions coloured green to indicate the acceptance tolerance, and yellow, orange and red, to indicate the size weight to be added.

Every engine produced by Ford Motor Company of America is tested for dynamic balance

Other signals to indicate engine speed and the position of unbalance at the front and rear of the engine are fed from the console to a stroboscopic unit mounted by the front engine pulley. This unit contains two stroboscopic lights—one triggered by a 1,500 c/min oscillator and the other by each vibration unbalance sensed by the pick-up.

A reference mark is placed on the accessory sheave before the engine enters the hot test area. After an engine has been connected on the hot test stand, the operator attaches the support arm containing the pick-ups to the conveyor carrier and the engine is throttled to 1,500 r.p.m. as indicated by the stroboscopic light. The operator then reads the meter, and if the unbalance does not exceed tolerance, no further work is required. If the unbalance does exceed tolerance at either the front, rear, or both, he engages the first light in the stroboscopic unit to observe the position of unbalance at the front and rear.

The engine is stopped and rotated manually to position the mark at the location observed under the stroboscopic light while the engine was running. An air tool is then employed to quickly drive a weight of proper size into pre-drilled holes in the flywheel or the front pulley—the size being indicated by the meters and the location by the stroboscopic light.

Correction

IN the 1957 Show Review issue of this journal reference was made, in the section dealing with "Brakes," to the Feeny and Johnson self-releasing, self-sealing, breakaway coupling for connecting the vacuum hoses of caravan or trailer brakes to the brake system of the tractor vehicle. In the same section there is an illustration of the Feeny and Johnson S.B.S. valve which is fitted on the trailer. Unfortunately, the wording of the caption to this illustration suggests that the breakaway coupling, and not the S.B.S. valve, is shown.

Recent Foreign Publications

Brief Reviews of Current Technical Books

Entwicklung von Verfahren und Geräten zur Prüfung Lichttechnischer Einrichtungen an Kraftfahrzeugen sowie Untersuchungen zur Ergänzung der bestehenden Prufvorschriften (Development of Procedures and Equipment for the Testing of Lamps and Illuminated Signals for Motor Vehicles, and also Research leading to the Completion of Test Specifications.)

In German. By Dr. Heinz Behrens
Düsseldorf: VDI-VERLAG G.M.B.H., 1957. 8¼ × 11¾. 30 pp. Price DM 13.50.

In this work, a series of lamp and signalling installations for motor vehicles and for use on the road is studied from the design point of view. As a result of this work, existing specifications to ensure efficient operation have been completed. The author states that for side lamps, an intensity of 0·5 candles is considered to be sufficient, since it cannot be distinguished from the dimmed headlamps from an adequate distance. For red tail lamps, the earlier specification of 0·2 candles appears to be too low. A value of certainly not less than 1 candle should be used. Red stop lamps should shine more intensely than the tail lamps, and even at values of 8 to 12 times that used for tail lamps, stop lamps are not adequate for all traffic conditions. Better results were obtained with a minimum light intensity of 50 candles for stop lamps and 10 candles for tail lamps.

In the report, it is suggested that there is no need to lay down a minimum size of tail lamp glass. It is stated that the checking of a tail lamp by comparing it with a standard form of lamp is unreliable and at the best can only serve as a means of deciding whether exact photometric measurements should be carried out.

Further investigations into usefulness and efficiency have been carried out on various warning devices, both for use on vehicles and to mark obstructions and danger points on roads. In the report, it is stated that flashing lamps are most effective as a warning device. The efficiency of reflectors can be assessed generally by the quality of the reflecting device. Their effectiveness as road signals undoubtedly is increased by placing several signals, adequate distances apart, ahead of the source of danger.

Hitherto, the effectiveness of number plate illumination lamps has been judged by the minimum brightness of the darkest spot and the brightness ratio between the brightest and darkest spots on the number plate. The distribution of brightness on the plate is of prime importance. A relative appraisal of this might be possible by the indication of either a brightness ratio between two adjacent points or a brightness gradient. However, at present, there is not sufficient test material available for a proper evaluation in this respect.

Untersuchungen über Mehrkreisige Druckmittelbremsen für Kraftfahrzeuge (Investigations of Multi-Circuit Pressure Actuated Brakes for Road Vehicles). Deutsche Kraftfahrforschung No. 102.

In German. By Hans Liebold, Dipl.-Ing.
Düsseldorf: VDI-VERLAG G.M.B.H. 1957. 11¾ × 8¼. 31 pp. Price DM. 13.90.

While pneumatic or hydraulic servo-actuated brakes have been developed to a high level of performance and reliability, their fundamental weakness is that, in case of damage of the supply line to a single wheel, the whole system might in certain circumstances become ineffective. The independent brake system usually provided in addition to the power-assisted brake is generally designed for parking only and, in any case, valuable time is lost before the driver can make use of it in an emergency. Because of this, the possibility of dividing the power brake into independent circuits to the different wheels has been considered from time to time. The monograph under review, No. 102 of the VDI series, covers an investigation to determine whether the development of such circuits should be fostered and, if so, how they should be arranged to obtain best results. As mentioned by the author, one of the earliest papers on this subject, by Bradley and Wood, was published in the January 1931 issue of *Automobile Engineer*.

In common with most other investigations of this type, the starting point is an assessment of the present state of development, coupled with a summary of operating experience with single and multi-circuit brake systems. Following this, the author briefly deals with the results of tests carried out with a lorry weighing 5·2 ton, laden. The tests were carried out as follows: with all wheels braked; front wheels braked; rear wheels braked; front left and rear right wheel braked; front right and rear left wheel braked; right front brake cut out of operation; left rear brake cut out of operation; and right rear brake cut out. To ascertain the effect of handed braking, the load was biased to one side.

Since tests of this nature are of general value only when correlated with and strengthened by theoretical considerations, the author considers the theoretical aspects of biased braking with particular reference to the steering and road-holding properties of the vehicle. This part of the work occupies the major portion of the monograph, and is also its most important and interesting part. It culminates in an example of the practical application of the theory worked out in considerable detail. The conclusions suggest that the adoption of multi-circuit brakes is to be favoured, and the monograph ends with an outline of requirements that should be met to ensure satisfactory performance. In common with previous papers, on the subject of vehicle braking, which have emanated from Prof. O. Bode's Road Vehicles Institute, at the Technical High School at Hanover, this publication is of a high standard and provides a wealth of valuable an thought-provoking information of considerable value to vehicle and brake designers.

Versuche an Zügen mit Selbsttatiger Lastanpassung der Anhängebremsen zur Ermittlung von Kenwerten für die Unmittenbare Bremsprüfung (Tests with Trains with Automatic Load Adjustment of Trailer Brakes to Determine the Characteristics of Direct Brake Tests). Deutche Kraftfahrforschung No. 101.

In German. By Prof. Dr. Otto Bode and Dipl.-Ing. Hans Liebold.
Düsseldorf: VDI-VERLAG G.M.B.H. 1957. 11¾ × 8¼. 21 pp. Price DM. 12.50.

This monograph is No. 101 of the road vehicle research series published by the German Society of Engineers. It deals with further work carried out at the Technical High School at Hanover on the subject of lorry and trailer brakes, previous work being reported in monographs Nos. 87, 89 and 97. According to the previous investigations, the determination of retardation alone is insufficient for the assessment of the function of a brake; more thorough methods are required to obtain better evaluation. Among these methods is the continuous recording of brake action, as well as of forces in the drawbar between the lorry and the trailer.

The latest report deals with the results of tests carried out with some of the more modern brake systems. In it, the authors consider the possibilities and difficulties of ensuring automatic regulation of the brake action, to suit the load requirements, with different brake types, such as air, hydraulic and inertia, or drawbar, types. They also discuss the requirements that must be met to ensure satisfactory operation. Extensive tests were carried out using a lorry weighing about 7 ton, fully laden, towing a 9 ton fully laden trailer. The trailer was tested equipped with, in turn, conventional pneumatic brakes, load-adjusted pneumatic brakes, hydraulic brakes, and drawbar-controlled penumatic brakes. All the test results were evaluated in terms of lorry, trailer and train performance. In addition, the shock encountered at the drawbar was determined as a function of the time interval between the start of brake pressure build-up and the attainment of full brake pressure. The magnitude of the maximum brake pressure was expressed as a function of the time required for brake application and of half the time for pressure build-up. Finally, suggestions are made for the testing of trailer brakes with automatic devices for their adjustment relative to the load carried.

In this latest VDI monograph, the high standards of the earlier issues are maintained. The work should be of particular interest and value to vehicle and brake designers, as well as to supervising authorities responsible for brake testing. It is well produced, and the numerous diagrams should be intelligible to engineers, even if they are not familiar with technical German. However, the actual subject matter covers only 15 pages, so the price works out at about 1s 3d per page, which is rather high.

Teoria, Konstruktia, Raschet i Ispitanie Dvigatelei Vnutrenego Sgorania (Theory, Design, Calculation and Testing of Internal Combustion Engines).

In Russian

Moscow: PUBLICATIONS OF THE ACADEMY OF SCIENCE. 1957. $10 \times 6\frac{1}{2}$. 201 pp. Price 11 roub 90 kop (10s).

The work under review is the third volume of research reports published by the Internal Combustion Engine Laboratory of the Russian Academy of Science. It opens with an eight page article, by Stechkin, Mihailov and Sviridov, commemorating the eightieth anniversary of Prof. Nicolai Romanovich Briling, head of the reciprocating engine division of the laboratory. Briling, whose teaching and design work dates back to 1907, and who, in conjunction with V. I. Grinevetzki is regarded as the originator of Russian compression ignition engine design and research, was responsible for the design and development of numerous engines mainly for transport applications and with outputs of up to 3,800 b.h.p. He has published numerous papers and several good text books, and his standing in Russia is equivalent to that of Sir Harry Ricardo in this country.

The first paper, by N. R. Briling, which occupies 30 pages, deals with the theory of the short-stroke diesel engine. In it, the author considers engines with stroke : bore ratios from 1·2 : 1, which is regarded as normal, to 0·8 : 1. First, he discusses the heat transfer problems to the coolant, then the starting characteristics, the peculiarities of the combustion process of the short-stroke engine, the effect of speed on cylinder charge, the effect of speed on inertia forces and dynamic loads, and the effect of stroke : bore ratio on thermal stresses and the specific output. This work is, perhaps, the most thorough and rigorous investigation so far published on this topical subject, and it is backed by some experimental results. From it, the author draws the following conclusions:

1. Reduction of the stroke : bore ratio increases the specific output and reduces specific weight. Thus, if for a conventional engine the specific weight amounts of 5·7 kg/b.h.p., then, for a short-stroke engine, it can be reduced to about 4·5 kg/b.h.p.
2. Similarly, a reduction of the stroke : bore ratio is of advantage for automobile engines, since it reduces the overall height.
3. The reduction of stroke : bore ratios is of particular benefit so far as the crankshaft rigidity and bearing reliability are concerned: increased crankshaft rigidity, and the resultant reduction in deflection, leads to improved lubrication. Thus, it is concluded that reduction of the stroke : bore ratio substantially improves engine design parameters.

Investigations of the flow conditions in gas turbine combustion chambers are dealt with in 20 pages by Prof. A. I. Mihailov. The theoretical aspects of the flow distribution are considered, together with an appreciable amount of experimental evidence. A method based on these considerations, is suggested for the calculation of the flow velocity distribution at various points of the chamber. This method is also suitable for the calculation of temperature and fuel density distribution.

The theoretical considerations and experimental data relating to the determination of the specific heat of technical gases are dealt with in 18 pages by M. D. Apashev, with particular reference to the equations published in 1939 and 1948 by Vukalovich and Novikov. Next, the effects of combustion process parameters on the indicator diagram are considered by Y. B. Svidorov. This paper, which occupies 22 pages, discusses the subject with particular reference to the propagation velocity, its effect on pressure rise and the effect of that rise on the indicator diagram and the cycle efficiency. In conclusion, a practical example is fully worked out.

An eight page paper, by A. V. Rachinsky, discusses the effect of fuel in the charge of carburettor engines on both the flow into the cylinders and volumetric efficiency. Then, in a paper that covers 38 pages, K. A. Sharpov considers, mainly against the background of American engine data, the possibilities of standardizing transport engines. A short paper, nine pages, by B. S. Stechin and M. D. Apashev, deals with tests carried out to ascertain simultaneously the flame propagation and pressure changes in a spark ignition engine. The thermodynamic analysis of the engine combustion process is dealt with in some detail, in 30 pages, by Y. B. Svidorov. He limits his investigations to the determination of the relationships between burnt mass and volume, and the rise of pressure and the propagation of burnt charge, as well as the plot of temperature distribution in the charge at any given moment during combustion. This paper concludes with a worked out example illustrating the proposed method. Finally, a 12 page paper by A. P. Lebedinski deals with road tests of vehicles, with particular reference to towed-load trials, the load being provided by a second car of a type similar to that on test.

This most interesting collection of papers is of value to engine research workers. Prof. Briling's paper is particularly welcome, because of its clarification of the main aspects of the stroke : bore ratio problem and its influence on internal combustion engine design.

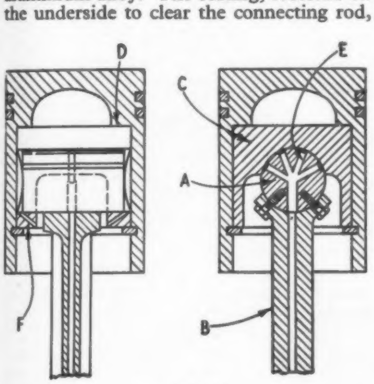
CURRENT PATENTS

A REVIEW OF RECENT AUTOMOBILE SPECIFICATIONS

Piston and connecting rod assembly

Gudgeon pin bearings in high performance engines are highly stressed mechanically and thermally and are of relatively restricted area. The attendant problems are largely met by the use of high-quality bearing materials and meticulous accuracy in machining and assembly. This design, intended particularly for two-stroke engines of the fuel-injection type, aims to provide a long-life, plain bearing using only a cheap bearing material.

The pin A is seated directly in the radiused end of the connecting rod B and secured by radially positioned set screws. An uninterrupted plain bearing for the pin is provided in a seating C of a suitable aluminium alloy. The seating, recessed on the underside to clear the connecting rod,



No. 769352

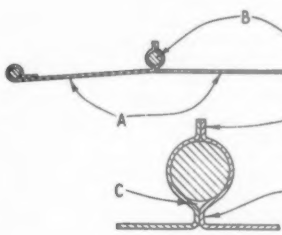
is cylindrical in shape and is fitted into the piston to abut a shoulder D. A spring ring retains it in its assembled position.

Lubricating oil is fed to the bearing by way of the drilled connecting rod and radial ducts E in the pin. Grooves for the distribution of the oil are machined longitudinally in the surface of the pin. On each side of the seating drillings F are provided for the return of oil to the sump. *Patent No.* 769352. *Klöckner-Humboldt-Deutz A.G.* (*Germany*).

Head lining

It is known to produce a head lining for a vehicle by sewing together strips of lining material with at least two rows of stitching suitably spaced apart to define a transverse pocket through which a listing rod can be passed. The invention provides an improved method of construction on those lines.

The lining is built up from a number of shaped strips A of a material comprising a textile base coated with polyvinyl chloride. It is secured in position in the vehicle by means of listing rods B passing through pockets C. To form these pockets, marginal portions of adjacent strips are folded upwardly so that the coated surfaces are face-to-face. Progressively the strips are joined by welding between high-frequency electrodes or steam-heated dies in a press. As convenient, welds D and E



No. 769449

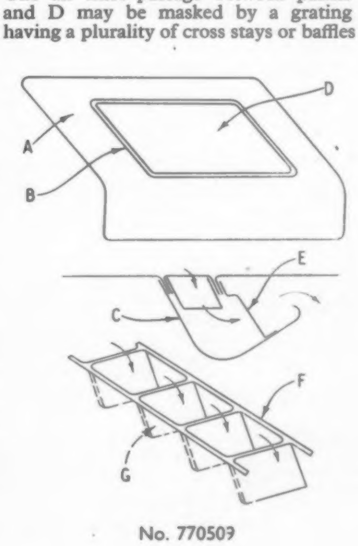
may be produced individually or simultaneously. In some instances it may be necessary to weld only half the length of the marginal portions at a time.

If the lining is held in position along its side edges by means of a piping enclosing a securing cord, the piping may be of similar material and also welded in position. *Patent No.* 769449. *Fisher and Ludlow Ltd.*

Cooling air inlet

Particularly in the case of rear-engined vehicles, difficulties are likely to be encountered in providing a cooling air inlet that is aesthetically pleasing yet not uneconomical in weight or cost. In the proposed construction the inlet offers the minimum interference with body line or air flow and the water trap serves as a structural reinforcement. The example shows the inlet incorporated in a sheet-metal panel which may serve as an engine inspection cover.

Panel A is formed with an aperture B of any suitable shape, the margin being turned inwardly to present a flange to which a water trap section C is attached. That part of the panel stamped out to form the aperture is flanged inwardly to form a complementary panel D and secured by a number of stays E to the water trap C. The air inlet passage between panels A and D may be masked by a grating F having a plurality of cross stays or baffles G

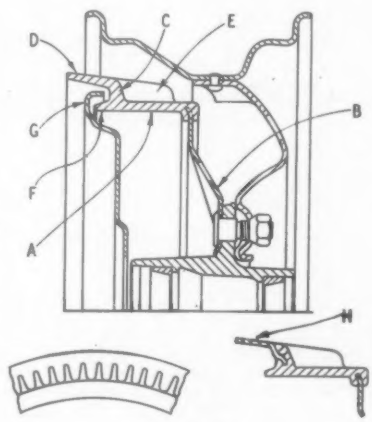


No. 770509

adapted to facilitate the entry of air and to influence the direction of flow. *Patent No.* 770509. *Volkswagenwerk G.m.b.H.* (*Germany*).

Brake drum

Improved heat dissipating characteristics and minimum distortion are claimed for this finned drum. The cast iron drum A is cast on the notched periphery of a pressed-steel head flange B and has near its open end a substantial radial flange C which is extended to form a flared annular circumferential rib D. On the exterior of the cylindrical portion of the drum are cast axially directed radial cooling fins E.



No. 771174

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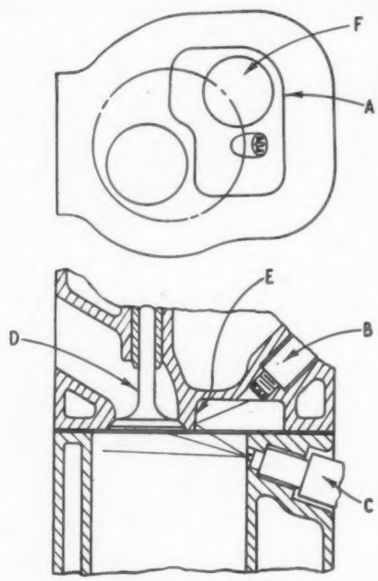

No. 768520

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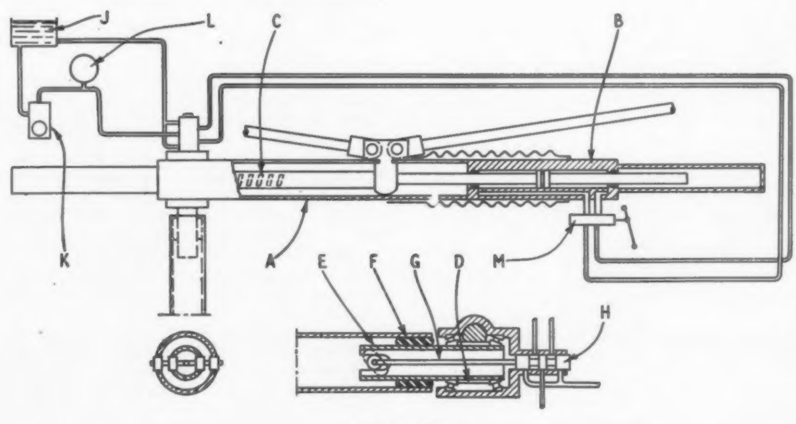

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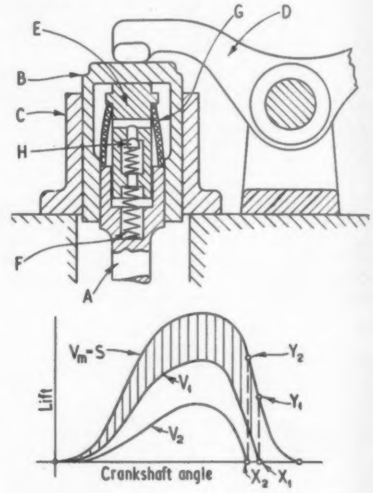

No. 768336

through a plunger B guided in a bushing C, a conventional overhead rocker lever D. A piston-type regulating member E is guided in the end of the pushrod and, under the constraint of spring F, holds plunger B up to the rocker end. The head of member E is sealed to the top of the pushrod by means of a resilient bellows G to form a reservoir for hydraulic fluid. A spring-loaded, non-return valve H is arranged in the interior of member E to control the passage connecting the pressure space, below member E, and the reservoir. When the device is subjected to axial pressure, either by the pushrod when the engine valve is being opened or by the rocker when the valve is being returned to its seat, the non-return valve closes and fluid can be transferred from pressure space to reservoir only by way of the clearance between member E and the pushrod guide at a predetermined leak rate. Between strokes the clearance between plunger B and rocker lever D is taken up by spring F; valve H opening and fluid being returned from the reservoir to the pressure space.

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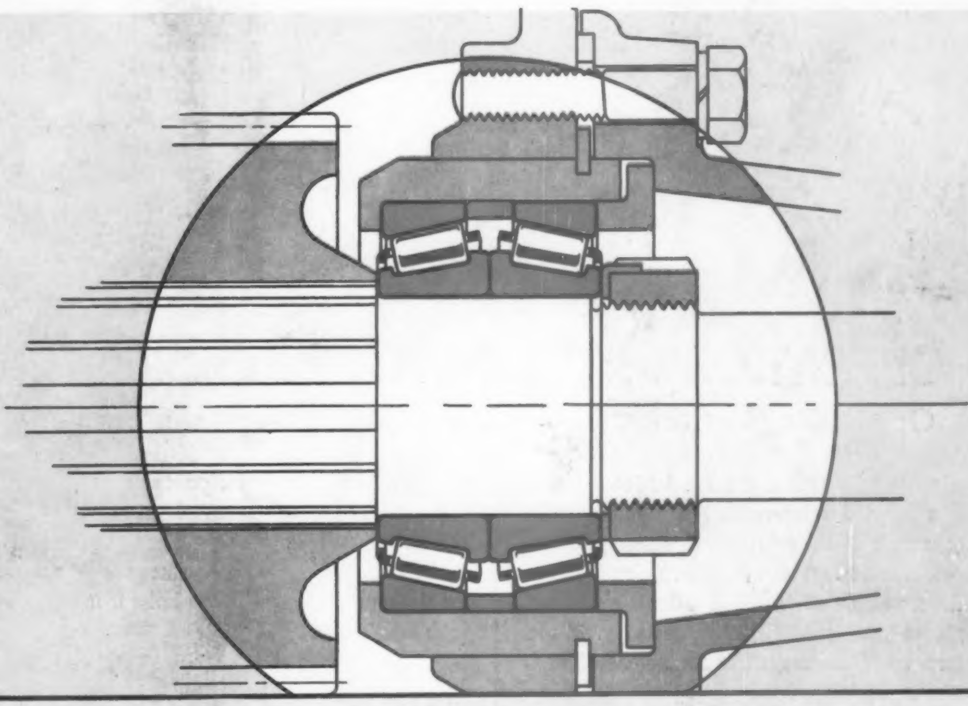
AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 48 No. 2 FEBRUARY 1958 PRICE: 3s. 6d.



In this latest VDI monograph, the high standards of the earlier issues are maintained. The work should be of particular interest and value to vehicle and brake designers, as well as to supervising authorities responsible for brake testing. It is well produced, and the numerous diagrams should be intelligible to engineers, even if they are not familiar with technical German. However, the actual subject matter covers only 15 pages, so the price works out at about 1s 3d per page, which is rather high.

Teoria, Konstruktia, Raschet i Ispitanie Dvigatelei Vnutrenego Sgorania (Theory, Design, Calculation and Testing of Internal Combustion Engines).

In Russian

Moscow: PUBLICATIONS OF THE ACADEMY OF SCIENCE. 1957. 10 × 6¼. 201 pp. Price 11 roub 90 kop (10s).

The work under review is the third volume of research reports published by the Internal Combustion Engine Laboratory of the Russian Academy of Science. It opens with an eight page article, by Stechkin, Mihailov and Sviridov, commemorating the eightieth anniversary of Prof. Nicolai Romanovich Briling, head of the reciprocating engine division of the laboratory. Briling, whose teaching and design work dates back to 1907, and who, in conjunction with V. I. Grinevetzki is regarded as the originator of Russian compression ignition engine design and research, was responsible for the design and development of numerous engines mainly for transport applications and with outputs of up to 3,800 b.h.p. He has published numerous papers and several good text books, and his standing in Russia is equivalent to that of Sir Harry Ricardo in this country.

The first paper, by N. R. Briling, which occupies 30 pages, deals with the theory of the short-stroke diesel engine. In it, the author considers engines with stroke : bore ratios from 1·2 : 1, which is regarded as normal, to 0·8 : 1. First, he discusses the heat transfer problems to the coolant, then the starting characteristics, the peculiarities of the combustion process of the short-stroke engine, the effect of speed on cylinder charge, the effect of speed on inertia forces and dynamic loads, and the effect of stroke : bore ratio on thermal stresses and the specific output. This work is, perhaps, the most thorough and rigorous investigation so far published on this topical subject, and it is backed by some experimental results. From it, the author draws the following conclusions:

1. Reduction of the stroke : bore ratio increases the specific output and reduces specific weight. Thus, if for a conventional engine the specific weight amounts of 5·7 kg/b.h.p., then, for a short-stroke engine, it can be reduced to about 4·5 kg/b.h.p.
2. Similarly, a reduction of the stroke : bore ratio is of advantage for automobile engines, since it reduces the overall height.
3. The reduction of stroke : bore ratios is of particular benefit so far as the crankshaft rigidity and bearing reliability are concerned: increased crankshaft rigidity, and the resultant reduction in deflection, leads to improved lubrication. Thus, it is concluded that reduction of the stroke : bore ratio substantially improves engine design parameters.

Investigations of the flow conditions in gas turbine combustion chambers are dealt with in 20 pages by Prof. A. I. Mihailov. The theoretical aspects of the flow distribution are considered, together with an appreciable amount of experimental evidence. A method based on these considerations, is suggested for the calculation of the flow velocity distribution at various points of the chamber. This method is also suitable for the calculation of temperature and fuel density distribution.

The theoretical considerations and experimental data relating to the determination of the specific heat of technical gases are dealt with in 18 pages by M. D. Apashev, with particular reference to the equations published in 1939 and 1948 by Vukalovich and Novikov. Next, the effects of combustion process parameters on the indicator diagram are considered by Y. B. Svidorov. This paper, which occupies 22 pages, discusses the subject with particular reference to the propagation velocity, its effect on pressure rise and the effect of that rise on the indicator diagram and the cycle efficiency. In conclusion, a practical example is fully worked out.

An eight page paper, by A. V. Rachinsky, discusses the effect of fuel in the charge of carburettor engines on both the flow into the cylinders and volumetric efficiency. Then, in a paper that covers 38 pages, K. A. Sharpov considers, mainly against the background of American engine data, the possibilities of standardizing transport engines. A short paper, nine pages, by B. S. Stechin and M. D. Apashev, deals with tests carried out to ascertain simultaneously the flame propagation and pressure changes in a spark ignition engine. The thermodynamic analysis of the engine combustion process is dealt with in some detail, in 30 pages, by Y. B. Svidorov. He limits his investigations to the determination of the relationships between burnt mass and volume, and the rise of pressure and the propagation of burnt charge, as well as the plot of temperature distribution in the charge at any given moment during combustion. This paper concludes with a worked out example illustrating the proposed method. Finally, a 12 page paper by A. P. Lebedinski deals with road tests of vehicles, with particular reference to towed-load trials, the load being provided by a second car of a type similar to that on test.

This most interesting collection of papers is of value to engine research workers. Prof. Briling's paper is particularly welcome, because of its clarification of the main aspects of the stroke : bore ratio problem and its influence on internal combustion engine design.

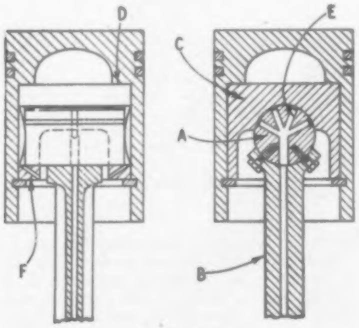
CURRENT PATENTS

A REVIEW OF RECENT AUTOMOBILE SPECIFICATIONS

Piston and connecting rod assembly

Gudgeon pin bearings in high performance engines are highly stressed mechanically and thermally and are of relatively restricted area. The attendant problems are largely met by the use of high-quality bearing materials and meticulous accuracy in machining and assembly. This design, intended particularly for two-stroke engines of the fuel-injection type, aims to provide a long-life, plain bearing using only a cheap bearing material.

The pin A is seated directly in the radiused end of the connecting rod B and secured by radially positioned set screws. An uninterrupted plain bearing for the pin is provided in a seating C of a suitable aluminium alloy. The seating, recessed on the underside to clear the connecting rod,



No. 769352

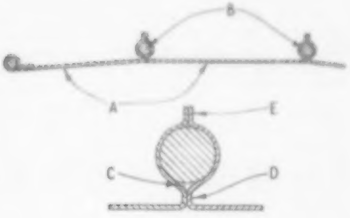
is cylindrical in shape and is fitted into the piston to abut a shoulder D. A spring ring retains it in its assembled position.

Lubricating oil is fed to the bearing by way of the drilled connecting rod and radial ducts E in the pin. Grooves for the distribution of the oil are machined longitudinally in the surface of the pin. On each side of the seating drillings F are provided for the return of oil to the sump. *Patent No.* 769352. *Klöckner-Humboldt-Deutz A.G.* (*Germany*).

Head lining

It is known to produce a head lining for a vehicle by sewing together strips of lining material with at least two rows of stitching suitably spaced apart to define a transverse pocket through which a listing rod can be passed. The invention provides an improved method of construction on those lines.

The lining is built up from a number of shaped strips A of a material comprising a textile base coated with polyvinyl chloride. It is secured in position in the vehicle by means of listing rods B passing through pockets C. To form these pockets, marginal portions of adjacent strips are folded upwardly so that the coated surfaces are face-to-face. Progressively the strips are joined by welding between high-frequency electrodes or steam-heated dies in a press. As convenient, welds D and E



No. 769449

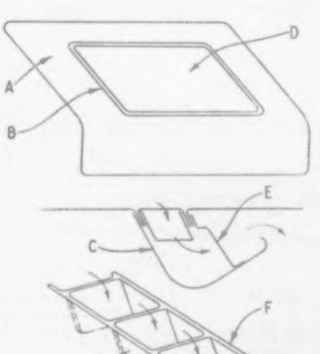
may be produced individually or simultaneously. In some instances it may be necessary to weld only half the length of the marginal portions at a time.

If the lining is held in position along its side edges by means of a piping enclosing a securing cord, the piping may be of similar material and also welded in position. *Patent No.* 769449. *Fisher and Ludlow Ltd.*

Cooling air inlet

Particularly in the case of rear-engined vehicles, difficulties are likely to be encountered in providing a cooling air inlet that is aesthetically pleasing yet not uneconomical in weight or cost. In the proposed construction the inlet offers the minimum interference with body line or air flow and the water trap serves as a structural reinforcement. The example shows the inlet incorporated in a sheet-metal panel which may serve as an engine inspection cover.

Panel A is formed with an aperture B of any suitable shape, the margin being turned inwardly to present a flange to which a water trap section C is attached. That part of the panel stamped out to form the aperture is flanged inwardly to form a complementary panel D and secured by a number of stays E to the water trap C. The air inlet passage between panels A and D may be masked by a grating F having a plurality of cross stays or baffles G



No. 770509

adapted to facilitate the entry of air and to influence the direction of flow. *Patent No.* 770509. *Volkswagenwerk G.m.b.H.* (*Germany*).

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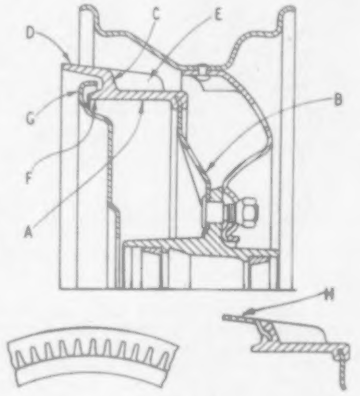

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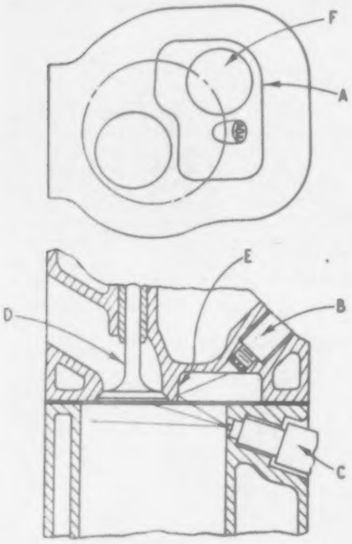

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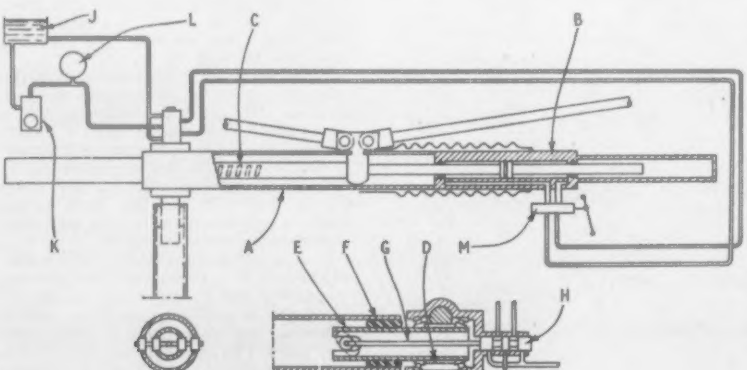

No. 768264

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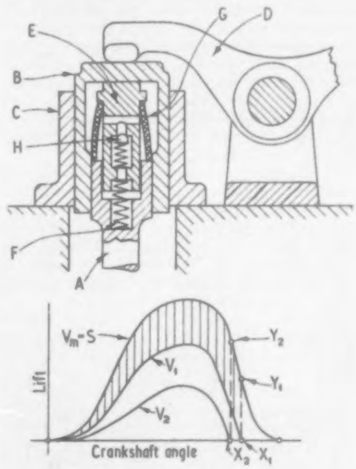

No. 768336

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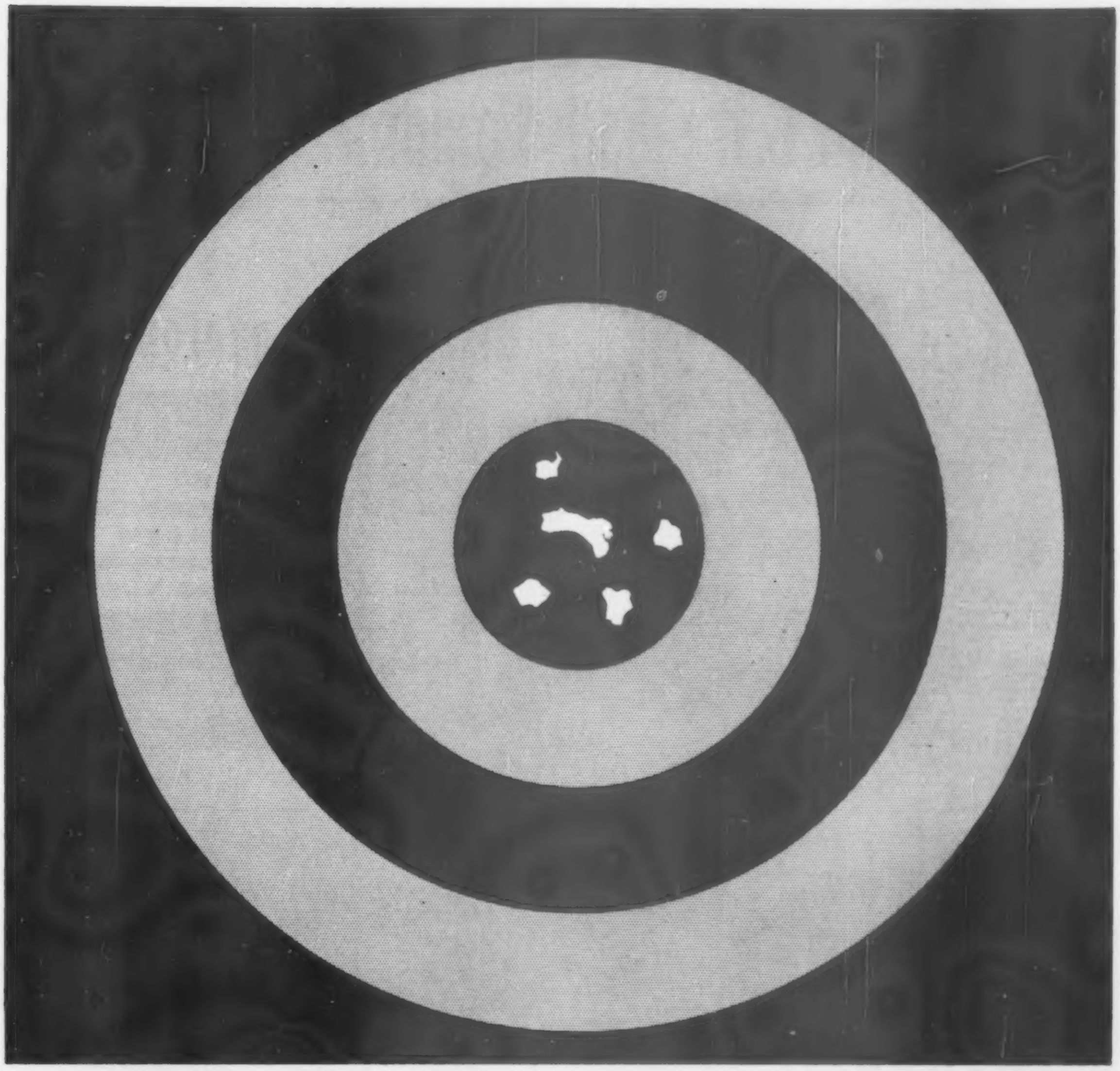

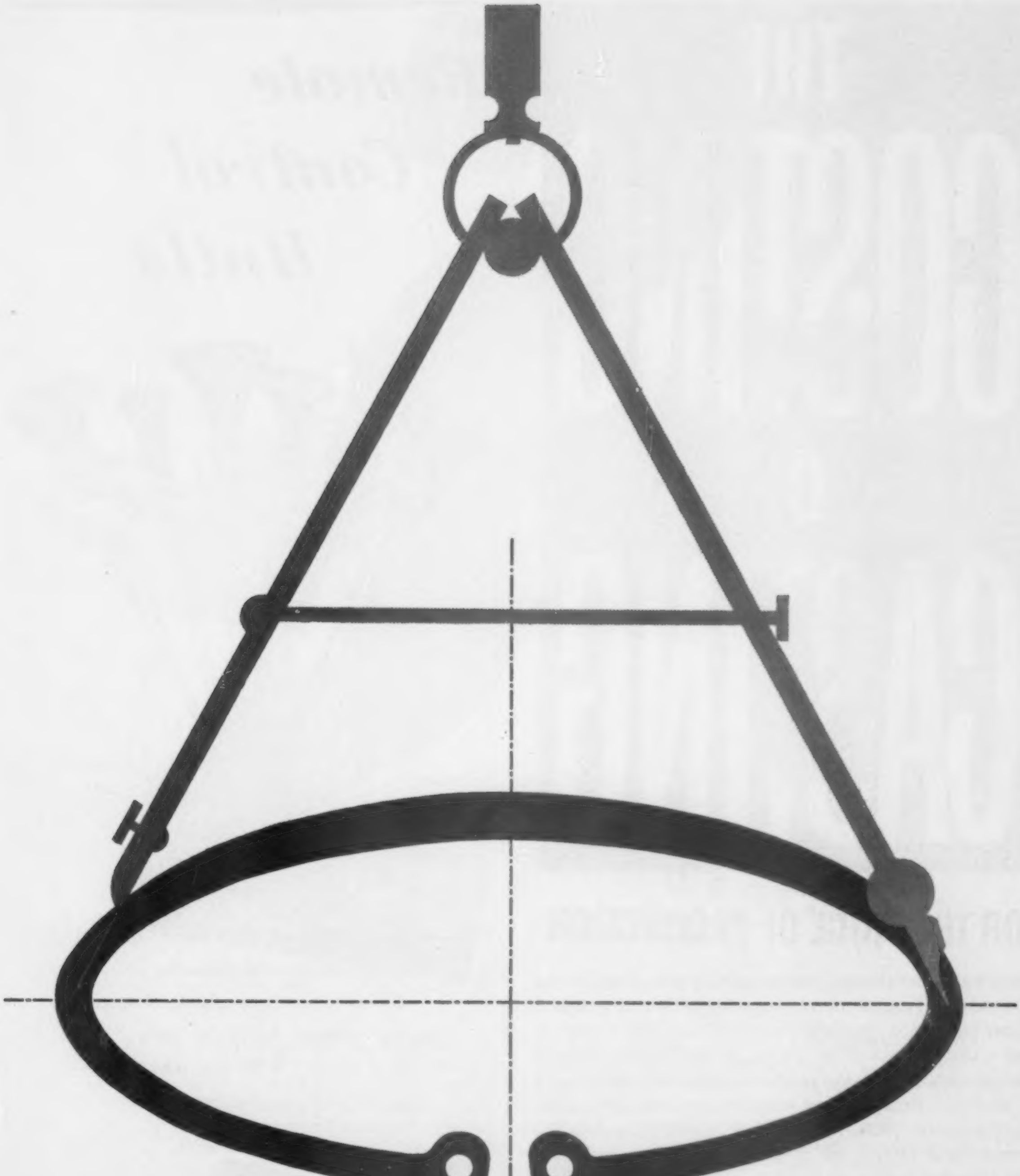

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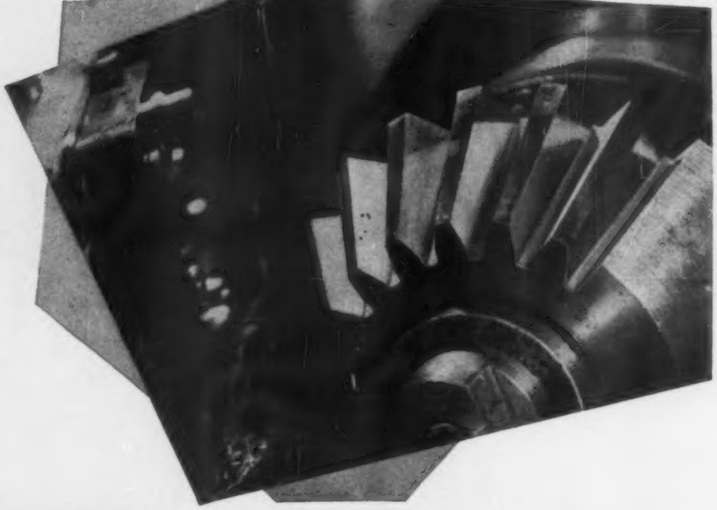

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4th	1.60/1	1.56/1	1.58/1	1/1
3rd	2.80/1	2.39/1	2.57/1	1.84/1
2nd	4.53/1	3.63/1	3.90/1	3.63/1
1st	8.14/1	6.51/1	7.00/1	6.52/1
Reverse	8.40/1	6.74/1	7.23/1	6.71/1

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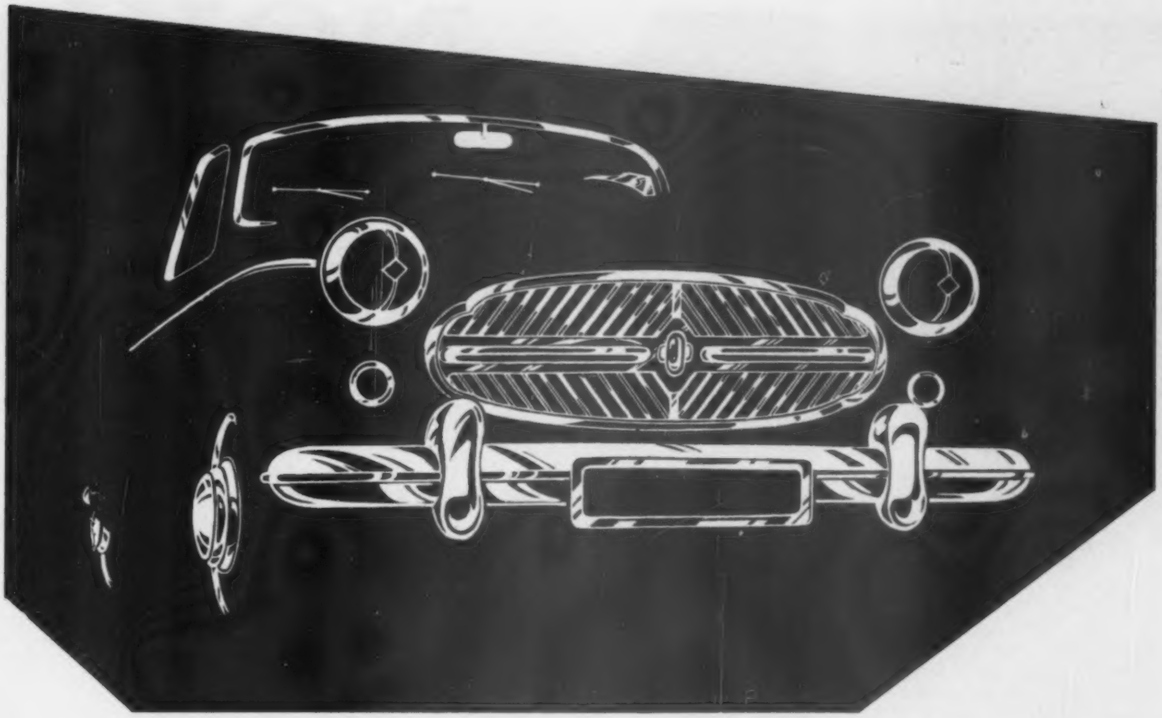

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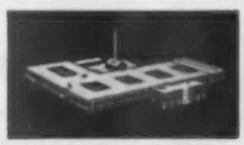

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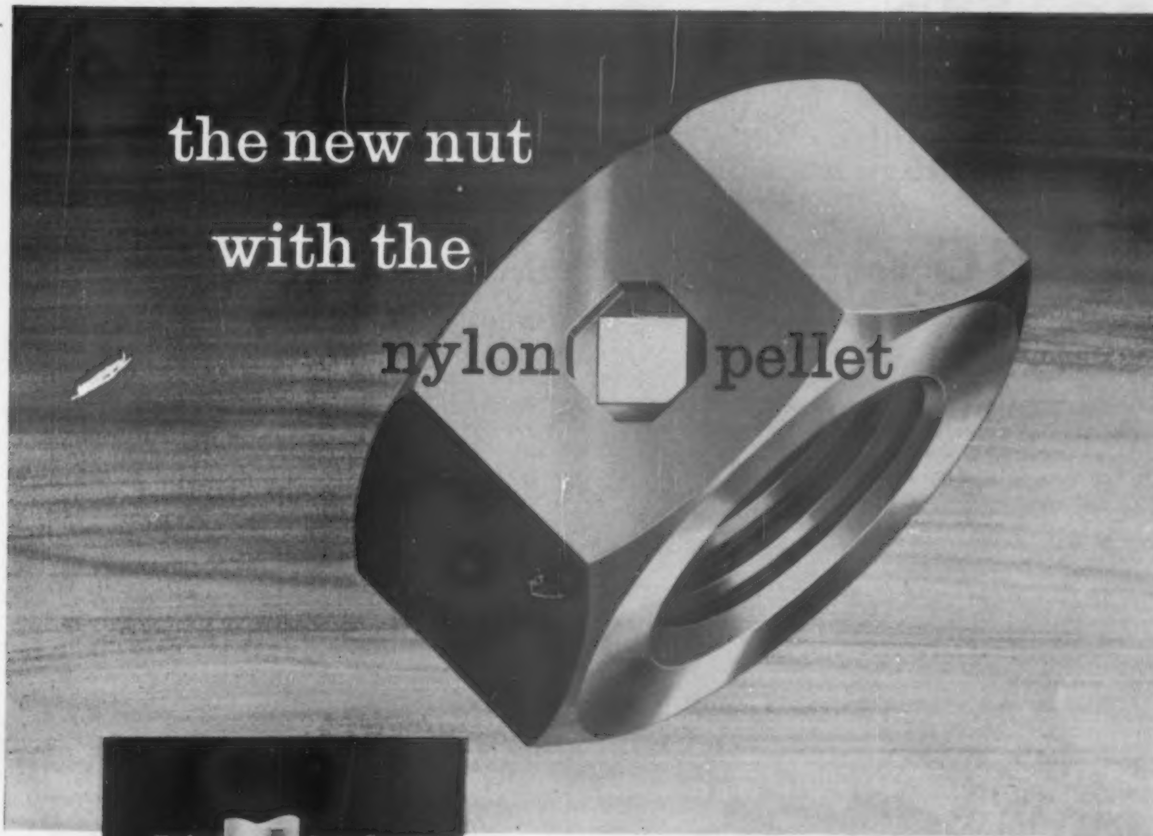

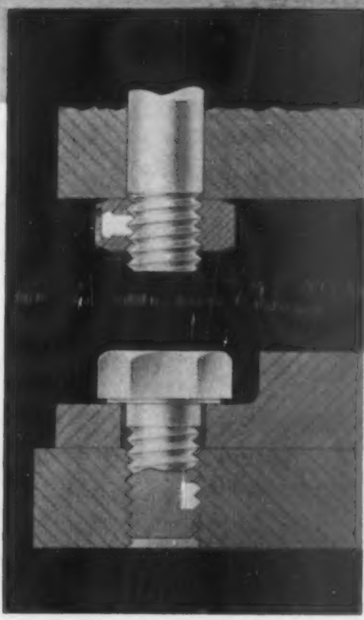

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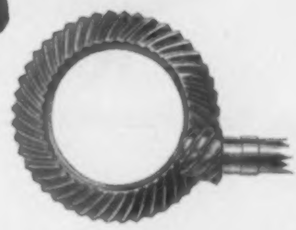

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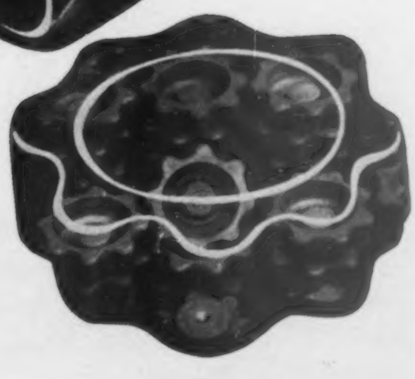

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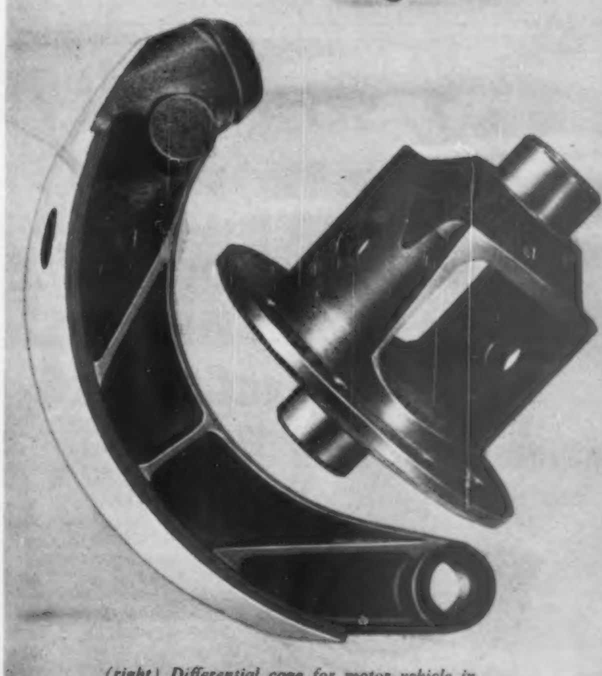

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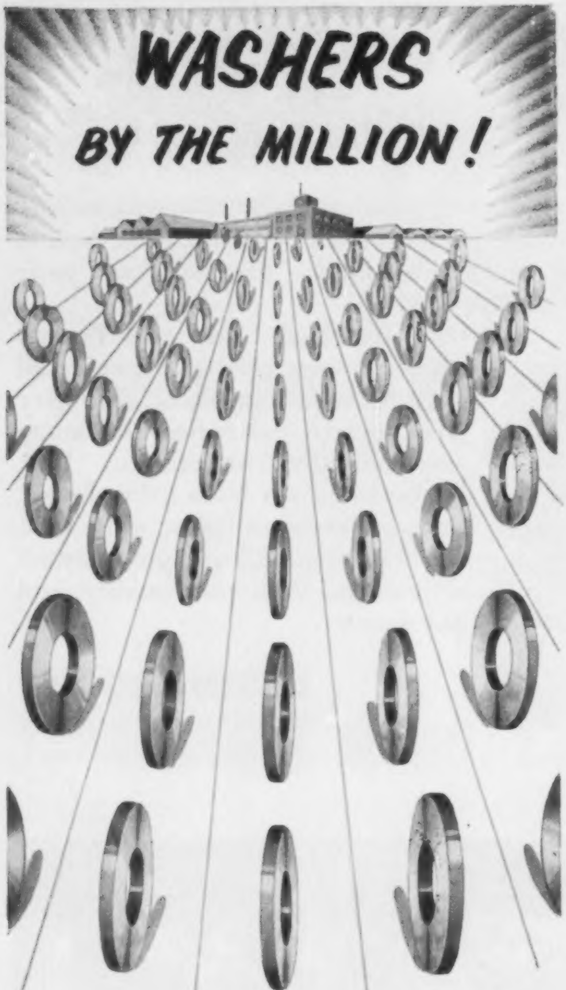

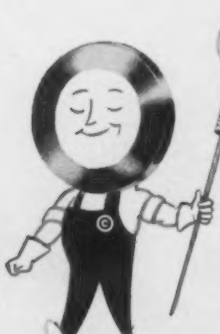

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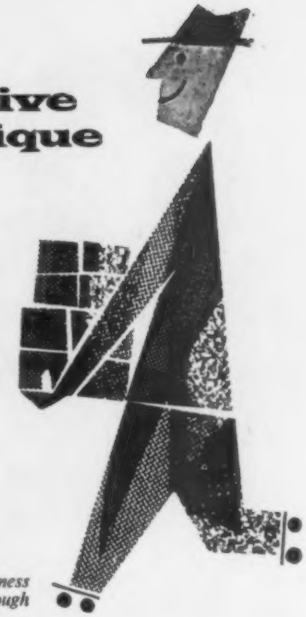

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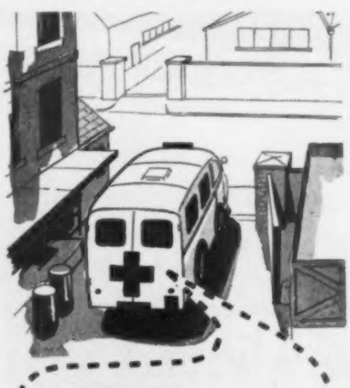

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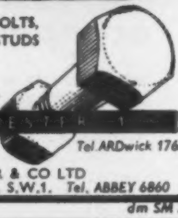

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